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FINAL REPORT
FOR STUDY OF NOZZLES
TO SPRAY CONTAMINATED FUEL

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Prepared By James P. Mitchell
James P. Mitchell

Approved By George D. Lewis
George D. Lewis

PRATT & WHITNEY AIRCRAFT
DIVISION OF UNITED AIRCRAFT CORPORATION
FLORIDA RESEARCH & DEVELOPMENT CENTER

FOREWORD

This is the Final Report for the program, the Study of Nozzles to Spray Contaminated Fuel and is submitted in accordance with the requirements of Work Statement Item 7 of Contract NOw61-0606-f. This report covers work done during the period 16 June 1961 to 26 December 1961.

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ABSTRACT

This report presents the results of a program that determined (1) the effects of contaminated fuel on the performance of turbojet main burner fuel nozzles and (2) the resulting effect on turbojet performance. Several types of nozzles were studied to determine the best method of spraying contaminated fuel. The fuel used in this program was contaminated with MIL-F-5007B contaminant, plus fungus. The fuel was filtered through a 150-micron filter.

The program consisted of the following parts: (1) an analytical study of the effects of contaminant accumulation on nozzle performance, (2) an analytical study of the effects of poor nozzle performance resulting from the use of contaminated fuel on turbojet performance, (3) nozzle spray tests in which several different nozzles were evaluated during cold-flow contaminated fuel spray tests, (4) burner can tests in which the effects of burning contaminated fuel in a conventional turbojet were determined, (5) a study in which the results of the burner can tests were used to determine the effects of burning contaminated fuel on turbojet performance, and (6) an analytical nozzle design study in which an air atomizing nozzle, an impingement nozzle, and two types of pressure atomizing nozzles were compared to determine which nozzle would be best for spraying contaminated fuel.

Results of the program disclosed:

1. Contaminant accumulation in the nozzle can distort uniformity of the spray being emitted from a nozzle and also change its flow schedule. These effects will result in poor temperature distribution and will cause hot spots at the burner can exit; which in turn can damage the turbine.

2. Contaminated fuel has little effect on large swirl type nozzles. Included in the Recommendations are the passage sizes required in a swirl type nozzle to spray contaminated fuel.

3. Air atomizing nozzles have the largest metering passages of the nozzles considered. Consequently, an investigation of the feasibility of using air atomizing nozzles in turbojets can be justified.

The conclusions of the program are presented at the beginning of each section of the report.

RECOMMENDATIONS

1. Because air atomizing nozzles have the largest metering passages of the nozzles considered, it is recommended that the feasibility of using air atomizing nozzles be investigated.

2. If conventional swirl type nozzles are to be used for spraying contaminated fuel, the nozzles should have the following passage sizes:

   a. The swirl slot dimensions should be at least 0.018 inch.

   b. The annulus thickness between the primary and secondary orifice in dual orifice nozzles should be at least 0.027 inch. Furthermore, the thickness should always be greater than the slot dimensions.

3. For nozzles with passage sizes as described in 2, it is recommended that large filters (approximately 350-micron size) be used at the nozzle inlet.

SECTION I

INTRODUCTION

It is impractical to keep turbojet fuel completely free of contamination, because of the equipment and procedures that would be required. Consequently, turbojet engines must always be capable of operation using fuel that contains a moderate amount of contamination. In addition, during emergencies or when engines are inadvertently supplied fuel that contains a large amount of contamination, they must continue to operate for a reasonable length of time (approximately 60 hours).

The critical engine part involved in ensuring safe operation with contaminated fuel is the fuel nozzle. When contamination accumulates in the metering parts of the nozzle, the flow schedule changes, and the spray pattern is affected. This can result in poor combustor performance, which in turn can result in reductions in thrust and increases in thrust specific fuel consumption.

The study covered in this report has determined the effects of contaminated fuel on the performance of fuel nozzles currently used on turbojet engines. Also, the study evaluated effects of poor nozzle performance resulting from the use of contaminated fuel on engine performance. In addition, a study was made to determine the best method of atomizing contaminated fuel.

For this study the turbojet fuel was contaminated according to the specifications in table I. It should be noted that these specifications are the same as MIL-E-5007B, except that the fungus solution has replaced the salt water. The fungus used during the program is defined in Appendix A.

The largest particle size specified in MIL-E-5007B is 200 microns. If this specification included only solid particles of contaminant (such as sand and dust), and if the fuel nozzles consisted of only straight circular passages, then passages as small as 200 microns (0.0079 inches) could be used. However, since fungus and lint are capable of bridging relatively large passages and since there are abrupt changes in flow direction and passage size in fuel nozzles, the minimum passage must be more than 200 microns. The determination of this minimum passage size was one of the objectives of this program.

The results of the study are presented in sections of this report as outlined below:

1. Contact with nozzle manufacturers
2. Analytical study of the effects of contaminated fuel on nozzle performance

3. General study of the effect of contaminated fuel on turbojet performance
4. Nozzle spray tests
5. Burner can tests
6. Specific study of the effect of contaminated fuel on turbojet performance
7. Nozzle design study.

TABLE I. CONTAMINANT SPECIFICATIONS

Contaminant	Particle Size	Quantity
Iron Oxide	0-5 microns	28.5 gm/1000 gals
	5-10 microns	1.5 gm/1000 gals
Sharp Silica Sand	40-50 mesh	1.0 gm/1000 gals
	50-100 mesh	1.0 gm/1000 gals
Arizona Road Dust	0-5 microns (12 percent)	8.0 gm/1000 gals
	5-10 microns (12 percent)	
	10-20 microns (14 percent)	
	20-40 microns (23 percent)	
	40-80 microns (30 percent)	
	80-200 microns (9 percent)	
U.S. Standard Staple No. 7 Prime Cotton Linters	As ground in a No. 7 Wiley mill and screened through a 4 mm screen.	0.1 gm/1000 gals
Crude Naphthenic Acid		0.03 percent by volume
Fungus Solution (10 parts salt water, prepared in accordance with MIL-E-5272 and 1 part fungus, as defined in Appendix A)		0.01 percent by volume

SECTION II

CONTACT WITH NOZZLE MANUFACTURERS

At the beginning of the study, the contaminated fuel nozzle program was discussed with several nozzle manufacturers, as required by Work Statement Item 2. The results of these contacts are summarized below:

1. Although some nozzles have been evaluated with contaminated fuel, none have been designed for spraying fuel contaminated with MIL-E-5007B contaminant plus fungus.
2. Several nozzles were recommended for evaluation during the comparative nozzle spray test.
3. Several reports covering contaminated fuel tests were received. In addition, the manufacturers furnished information on fuel nozzle theory.
4. Delavan Manufacturing Company was subcontracted to make the analytical studies of the effect of contaminated fuel on nozzle performance (Work Statement Item 1) and the analytical nozzle design studies (Work Statement Item 3). Reports covering Delavan's work were presented in Monthly Progress Reports 4 and 6 (PWA FR-[illegible]07 and PWA FR-337).

Appendix B includes a list of the manufacturers that were contacted, a list of the reports that were received, the results of these reports that were pertinent to this program, and a list of the nozzles that were recommended.

SECTION III

ANALYTICAL STUDY OF THE EFFECTS OF CONTAMINATED FUEL ON NOZZLE PERFORMANCE

A. GENERAL

1. *BACKGROUND*

Work Statement Item 1 of the contract required analytical studies of (1) the effects of contaminant accumulation on fuel nozzle performance and (2) the resulting effects on turbojet performance. Delavan Manufacturing Company was subcontracted to help in the analytical study of the effects of contamination on nozzle performance; a report (Delavan Manufacturing Company Technical Report No. 196) covering their work was presented in Monthly Progress Report No. 4 (Pratt & Whitney Aircraft Report No. PWA FR-297).

The results of the study of the effect of contaminant accumulation on nozzle performance are presented in this section. The results of the study of the effect of contamination on turbojet performance are presented in Sections IV and VII. In section IV, the results of a general consideration are presented; in section VII, the results of the burner can tests with contaminated fuel are used to determine the effects on engine performance.

2. *NOZZLE TYPES AND PERFORMANCE PARAMETERS*

The fuel nozzles that are currently used in the main burners of turbojets are all of the pressure atomizing type. These include the simplex, duplex, dual, spill or bypass, and the variable area nozzles. The characteristics of each of these nozzle types are presented in Appendix C.

The performance parameters that are used in evaluating nozzles are droplet size, spray angle, spray uniformity, flow range, and repeatability of fuel flow at a given pressure. These parameters are discussed below:

a. Droplet size is a measure of the degree of atomization achieved; this is an important parameter because it directly affects the design of the burner can. Sprays with mean droplet sizes in the 100 to 200 micron range are used in turbojets. (Because the nozzle sprays are made up of droplets of several sizes, mean droplet size is defined as the droplet size of a uniform spray containing the same fuel volume and the same number of droplets as the actual spray.)

b. Sprays with hollow spray cones and large spray angles are used in turbojets because these characteristics permit the fuel to be mixed with the air in a shorter length of the combustor. Spray angles in the 70 to 90 degree range are used. It is important to have atomizers that will maintain a given spray angle throughout the nozzle flow range.

c. Spray uniformity is desirable because in the burner can it will govern the local fuel-to-air mixture ratio. Spray uniformity may be determined experimentally by measuring the quantity of fuel in different sectors of the spray cone.

d. Because fuel flow varies widely in a turbojet, the nozzle must provide fuel sprays that have the characteristics discussed above over the entire flow range. Flow ranges with a maximum-to-minimum flow ratio of 30 to 40 are required in turbojets.

e. Flow repeatability is important because several fuel nozzles are operated in parallel in an engine. Since pressure drop across the nozzle determines the relative fuel distribution among nozzles, the fuel-to-air ratio in the burner can will not be uniform if there is hysteresis in the flow schedule of the nozzle, or if the nozzles are mismatched.

B. CONCLUSIONS

From the considerations presented later in this section on the analytical study of the effects of contaminant accumulation on nozzle performance, the following conclusions were made:

1. There are three major locations in a swirl nozzle where contaminant accumulation can produce poor nozzle performance. These are (1) the swirl slots, (2) the swirl chamber, and (3) the outlet orifice. Of these three locations, accumulation in the swirl slots is the most likely.

   a. Contaminant accumulation in the swirl slots will cause (1) a significant increase in nozzle pressure drop, (2) an increase in the spray angle, (3) a reduction in mean droplet size, and (4) a reduction in spray uniformity. Of these, the most severe effect is the increase in nozzle pressure drop.

   b. Contaminant accumulation in the swirl chamber and at the orifice outlet can cause (1) poor spray uniformity with streaks or a skewed spray and (2) an increase in the nozzle pressure drop. Both of these could cause detrimental effects in an engine.

2. In the commonly used dual orifice nozzle the contaminant can collect in the small annulus between the primary orifice and secondary orifice and cause streaks in the spray pattern.

3. Contaminant accumulation in the flow divider valve can cause hysteresis in the flow schedule and can result in poor fuel distribution in the burner can.

4. The effects of contaminated fuel on the variable area nozzle are similar to those on the flow divider valve.

C. DISCUSSION

The effects of contaminant accumulation are discussed below. The approach taken was to determine the effects on the performance of a nozzle that is representative of the size and type that is used in current turbojets. Specifically, the simplex nozzle sized for a flow rate of 100 lb/hr was studied. Following a consideration of this simplex nozzle, assessment of the effects is given to other nozzle types. The results of these considerations are presented below.

1. THE SIMPLEX NOZZLE

All of the swirl type nozzles (dual, duplex, and spill) are derived from the simplex nozzle. Consequently, the simplex nozzle can be used to illustrate the general effects of contaminant on nozzle performance. Figure 1 shows a typical simplex nozzle. The critical dimensions of this nozzle are indicated in the figure; these include the swirl slots, the swirl chamber, and the outlet orifice.

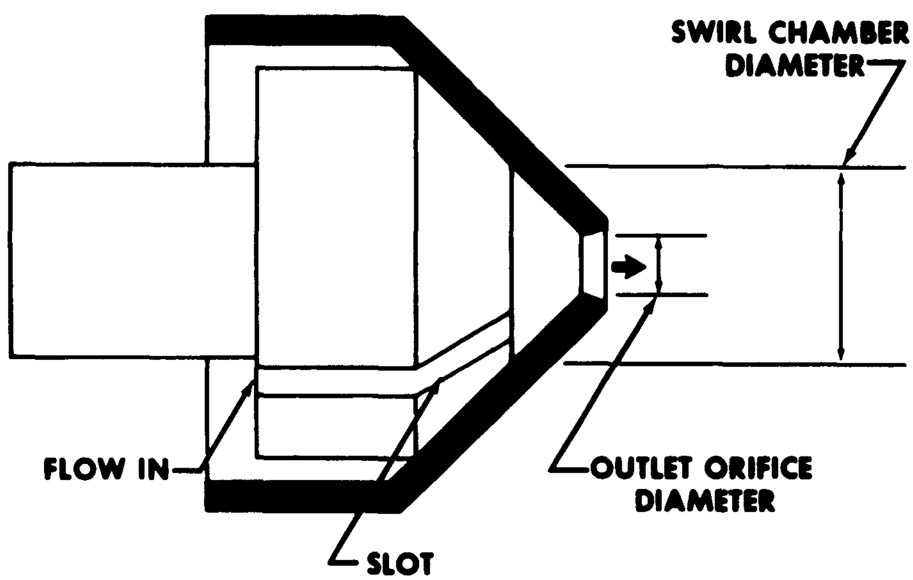


Figure 1 Simplex Nozzle Schematic FD 3433

The size of the passages depends on the flow rate for which the nozzle is designed. In figure 2, the size of the critical dimensions and the nozzle pressure requirements are presented as a function of flow rate. As shown in the figure, one of the large 200 micron particles could block a swirl slot of a nozzle that is designed to operate in the 0 to 10 lb/hr flow range and that has eight swirl slots. (The swirl slot size is a function of the number of slots, which is selected based on nozzle performance; the spray uniformity increases as the number of slots increases. Normally between one and eight slots are used.) For nozzles sized for flows above this range, the contaminant would have to build up before a slot could be plugged. Furthermore, if there were only solid particle contaminants, this build up might

not occur; however, cotton linters and fungus contribute to the accumulation since the linters can bridge openings, and since the fungus will stick to metal surfaces.

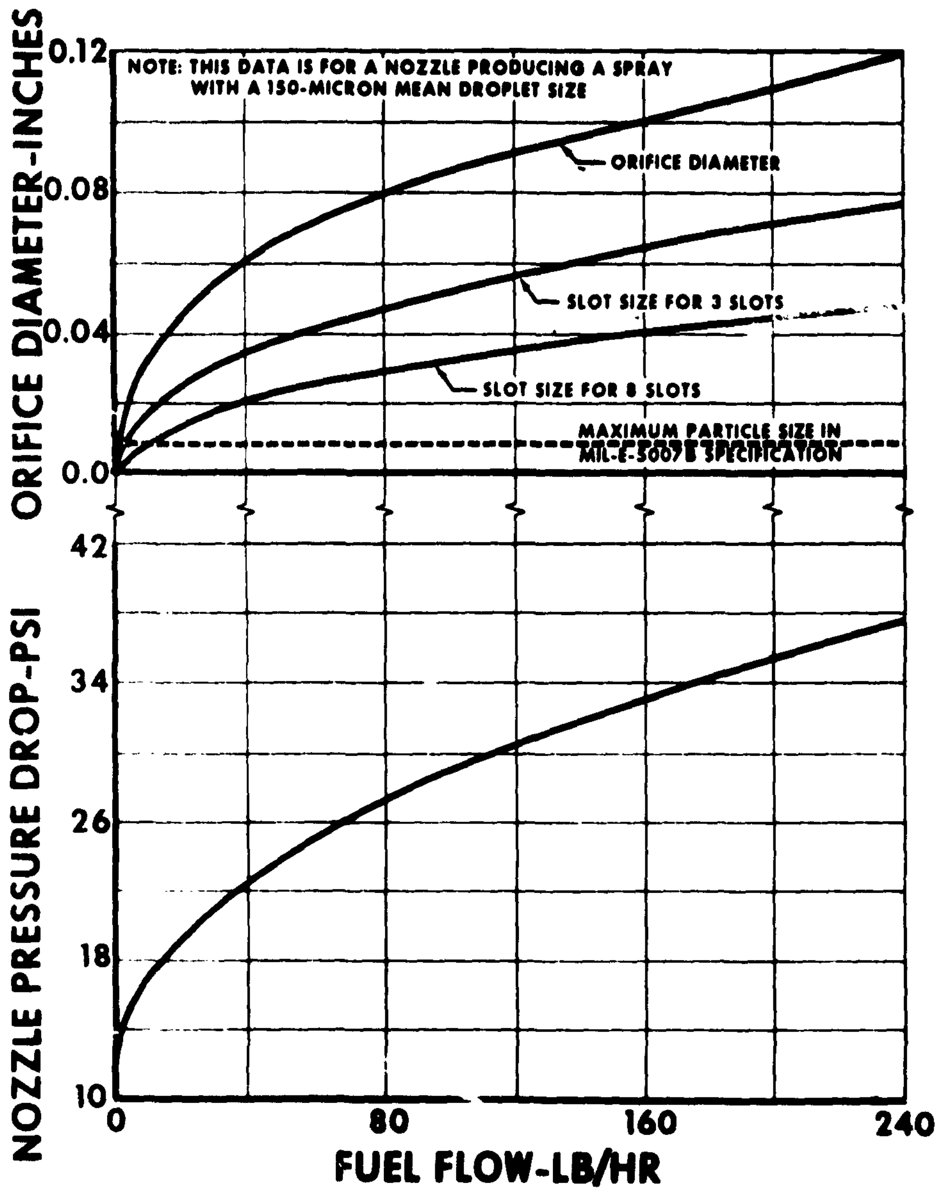


Figure 2 Simplex Nozzle Dimensions and Pressure Requirements *FD 3526*

To quantitatively assess the effects of contamination on nozzle performance, a nozzle designed to produce a spray with a 150-micron mean droplet diameter and a spray angle of 80 degrees at a flow rate of 100 lb hr was selected for consideration. The effects of plugging the swirl slots, contaminant accumulation in the swirl chamber, and a plugged outlet orifice were determined; the results are presented below:

a. Effects of Plugged Swirl Slots – The most detrimental effect of plugging a slot is the change in flow schedule. This will result in poor fuel distribution in the burner can. This point is il-

lustrated in figures 3 and 4, which show the effect of plugged slots on pressure drop, spray angle, and droplet size. (It should be noted that all these curves are based on the assumption that the same fuel flow would be flowing through the nozzle.) As

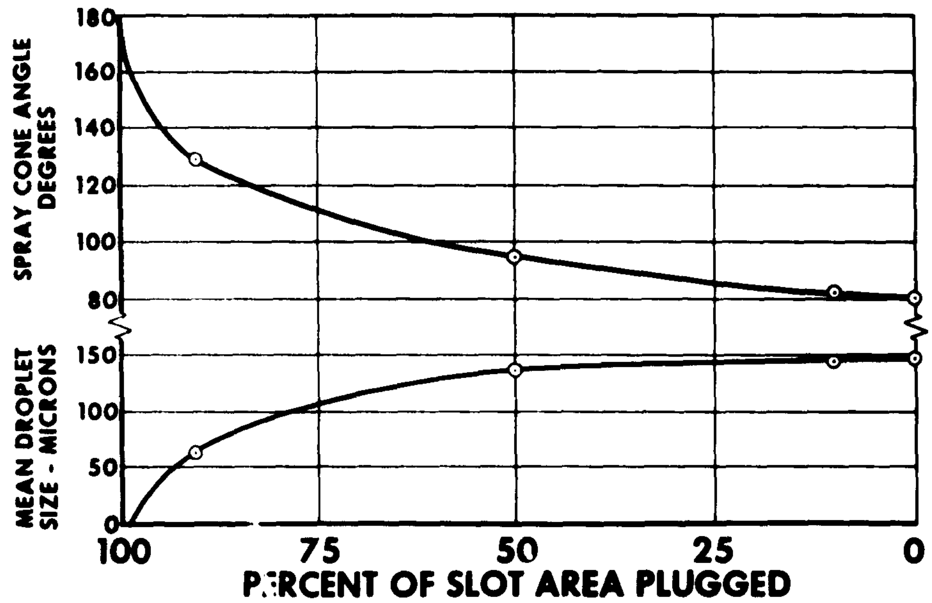


Figure 3 Effect of Plugged Swirl Slots on Spray Angle and Droplet Size *FD 3499*

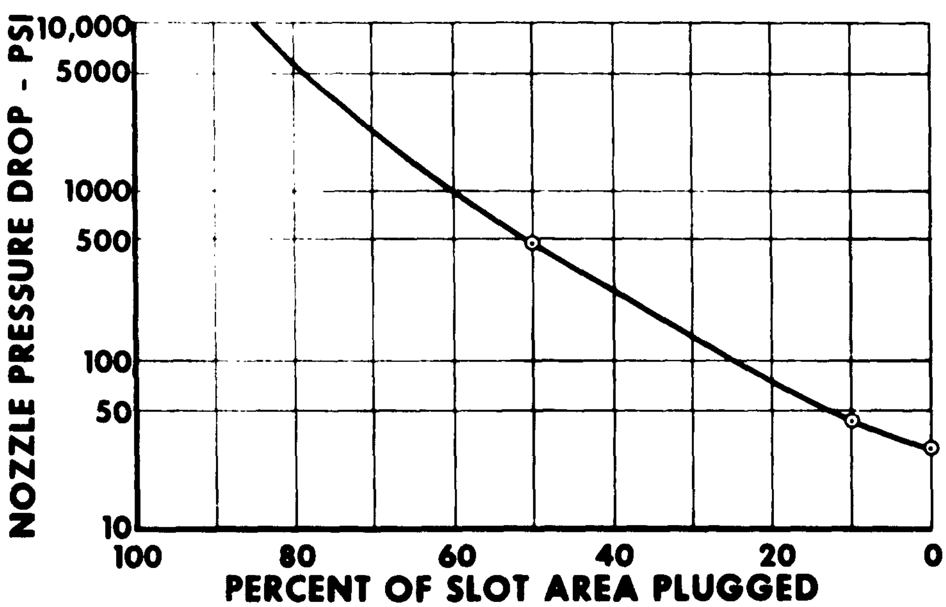


Figure 4 Effect of Plugged Swirl Slots on Nozzle Pressure Drop *FD 3502*

an example of the effect on performance, assume that the nozzle contains four swirl slots that are 50 percent plugged. (Either two of the slots are completely plugged and the other two are clean, or 50 percent of the total area of the four slots is plugged with contaminant.) For this case, the mean droplet size will decrease from 150 to 135 microns, the spray angle will increase from 80 to 95 degrees, and the pressure drop will increase from 28 to 450 psi. For other percentages of slot area stoppage, the effect on droplet size, spray angle and nozzle pressure can be obtained from the curves.

In addition to these effects, plugged slots will result in a spray with poor uniformity. If all of the slots are plugged the same amount the spray uniformity theoretically would not change. However, if one or two of the slots are plugged and the others remain clean the uniformity will decrease.

b. Effect of Contamination in the Outlet Orifice – The effects of partially plugging the outlet orifice were calculated. The results of these calculations are shown in figures 5 and 6. As an illustration of the effects of a plugged outlet orifice, if the nozzle is 50 percent plugged, the mean droplet size will decrease from 150 to 98 microns, the spray angle will decrease from 80 to 68 degrees and the pressure drop will increase from 28 to 115 psi.

The effects predicted on spray angle and droplet size are for a circular orifice. Therefore, the results are those for the condition when a uniform buildup of contaminant occurs on all sides of the orifice. This condition is not as severe as the more likely condition – when the outlet orifice is not circular; that is, when contaminant accumulates at one side of the orifice and causes the fuel to be emitted from a portion of the orifice. This condition can cause severe streaks in the spray pattern. As an example of this effect, the curves shown in figure 7 are presented. These curves were presented by Tate and Marshall[1]. The curves show the flow distribution obtained with a circular orifice and that obtained with a slightly noncircular orifice. The nonuniformity caused by the latter is evident.

The results presented for the pressure drop will hold regardless of the shape of the orifice.

[1]Tate, R. W. and Marshall, W. R., "Atomization by Centrifugal Pressure Nozzles," Chemical Engineering Progress, May, 1953

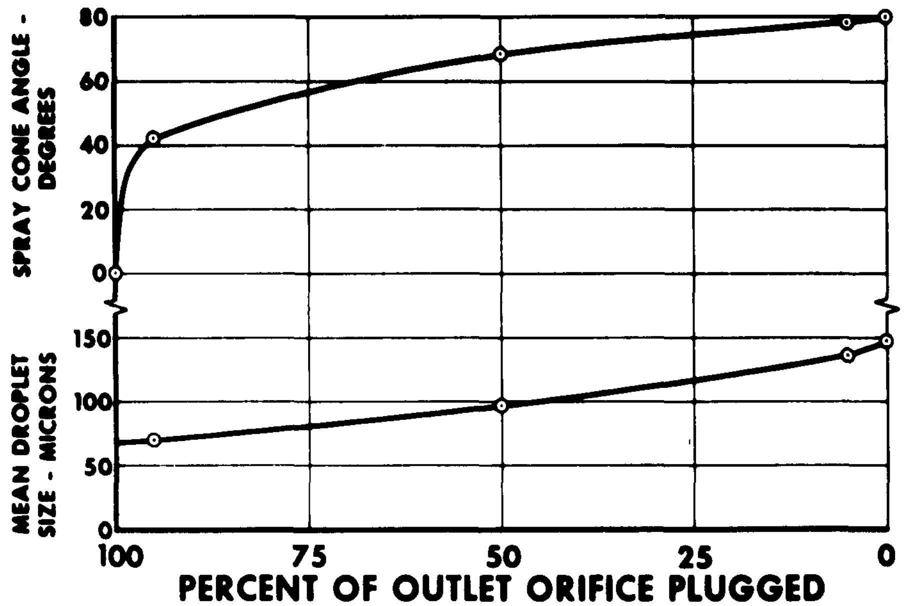


Figure 5 Effect of a Plugged Outlet Orifice on Spray Angle and Droplet Size FD 3501

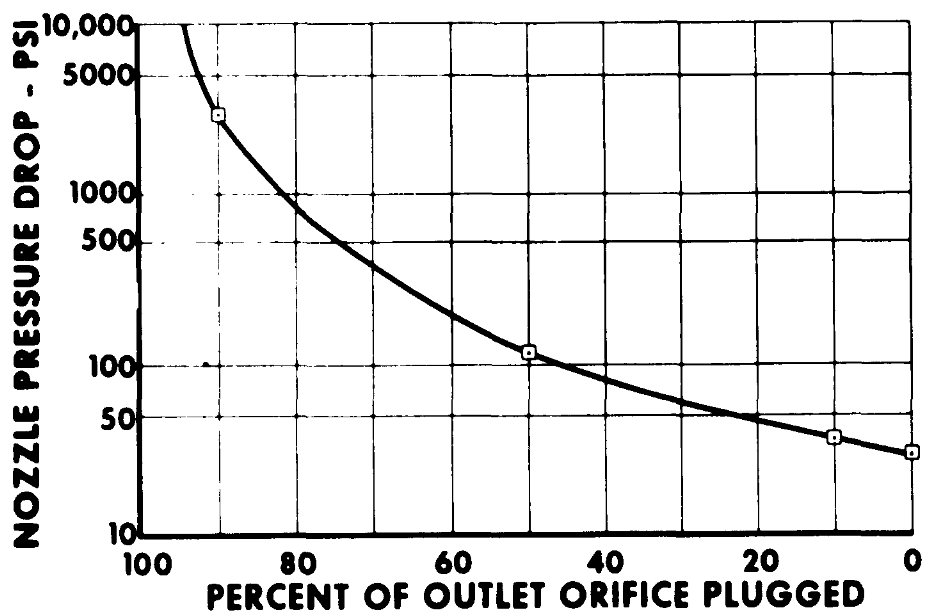


Figure 6 Effect of a Plugged Outlet Orifice on Nozzle Pressure Drop FD 3503

2. OTHER PRESSURE ATOMIZING NOZZLES

The performance of the other swirl type nozzles will be affected in a manner similar to the simplex. The dual orifice nozzle that is used on most turbojets has one contaminant accumulation location that the simplex nozzle does not have. The annulus between the primary and secondary orifices in some dual orifice nozzles is as small as 0.01 inches. This location is illustrated in figure 8. If contaminant accumulates in the annulus, the effects will be the same as those described for the plugging of the outlet orifice of a simplex nozzle. The most severe of these effects are heavy streaks and a skewed spray pattern.

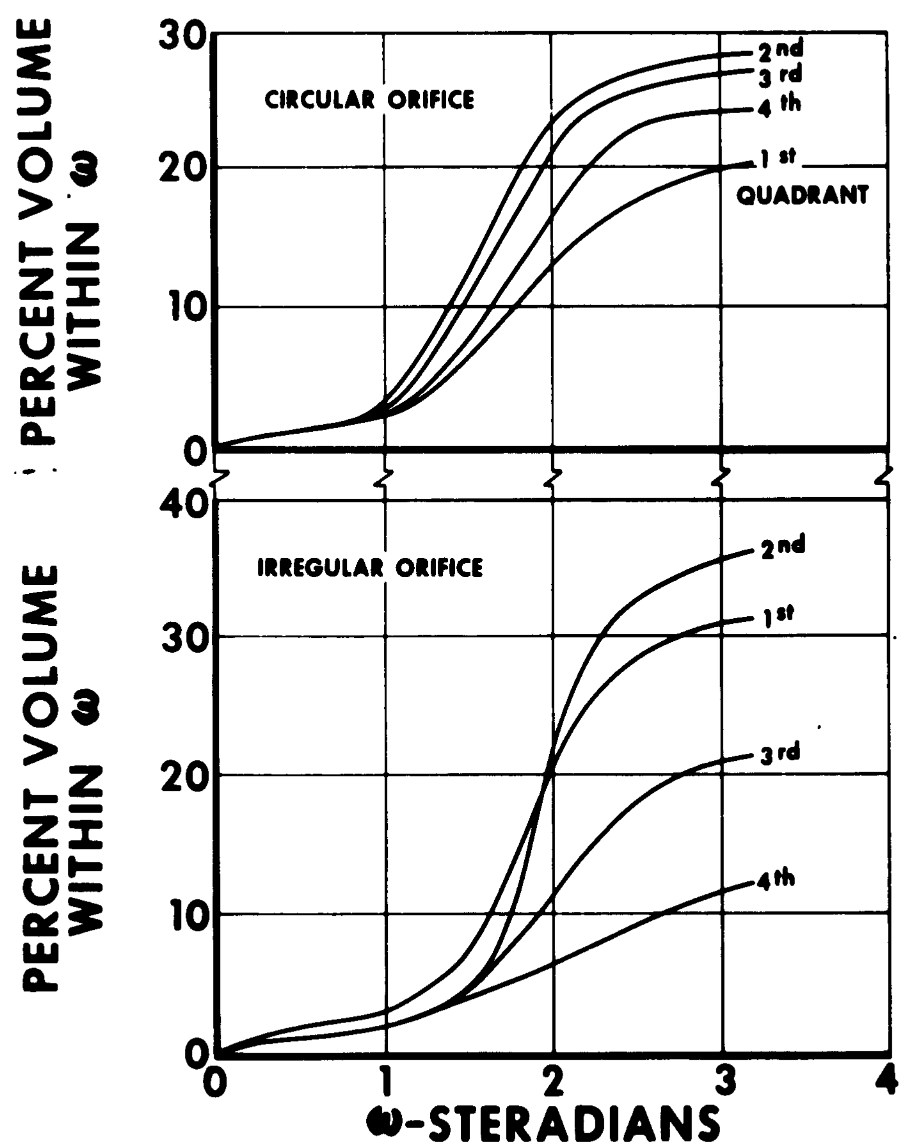


Figure 7 Effect of Orifice Shape on Spray Uniformity FD 3519

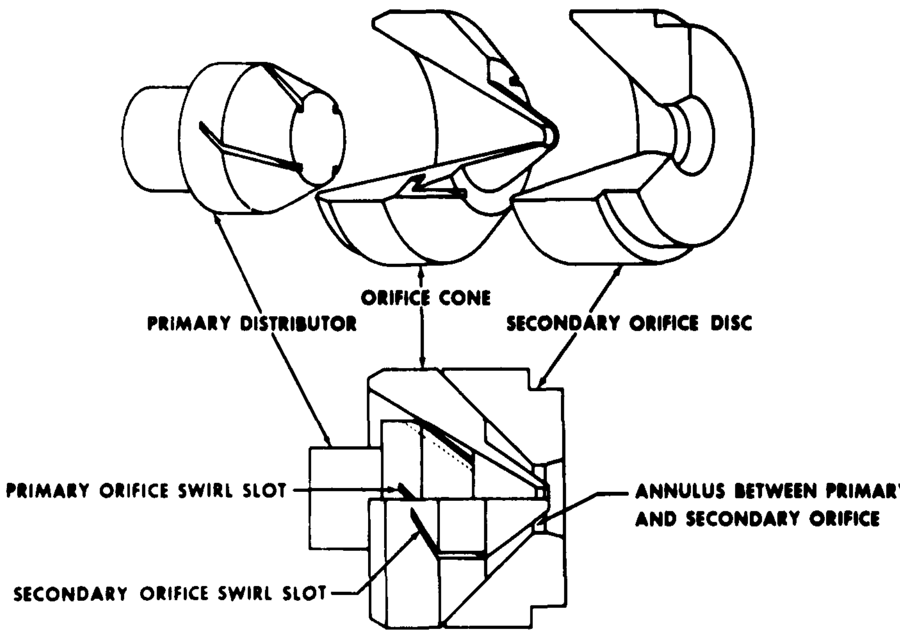


Figure 8 Dual Orifice Nozzle Schematic *FD 3560A*

3. *FLOW DIVIDER VALVE*

Because the pressure requirements to obtain good atomization over the entire flow range of a turbojet are excessive for a simplex nozzle, a dual orifice nozzle system is normally used. In this system, a small simplex nozzle (primary orifice) provides atomization at low flow rates, and a large simplex nozzle (usually one that houses the primary and that is called the secondary orifice) is used to provide atomization at high fuel flows. Normally, a flow divider valve that is actuated by pressure is used to schedule flow through the secondary orifice.

A typical flow divider valve is shown in figure 9. The spring keeps the poppet or pintle against its seat until the pressure reaches a predetermined value, where the poppet is lifted from its seat and flow is directed through the secondary orifice.

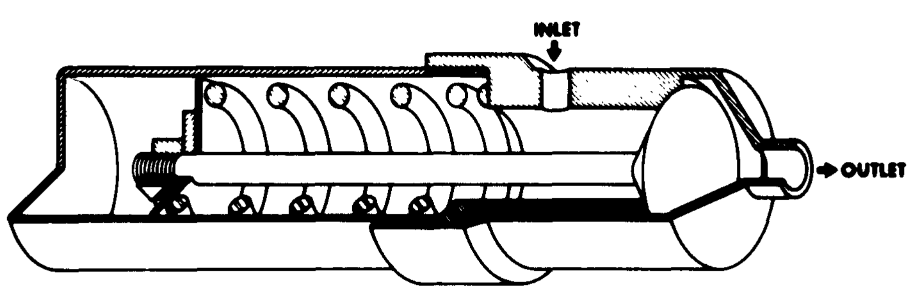


Figure 9 Flow Divider Valve Schematic ***FD 3557***

The only critical location in the valve where contamination can be a problem is at the poppet seat. Valve size determines the susceptibility of a flow divider valve to contaminant accumulation. In figure 10, the valve seat lift is presented for two different valves that are designed according to the data presented. The curves indicate that for small valves the lift could be less than 200 microns (the size of the largest contaminant particle) through a large portion of the flow range. When operating at these low flows, contaminants will accumulate behind the poppet. This will increase the pressure drop across the valve, causing the valve lift to increase. As a result a flow divider valve could be self-cleaning, at least for small quantities of accumulation. This characteristic should not result in poor atomizer performance.

The desirability of having a central flow divider valve supplying several nozzles rather than one valve for only one nozzle is obvious from the data presented in figure 10. This point is made clear by a comparison of the variation in valve lift with flow for valves 1 and 2. Since valve 2 is eight times larger than valve 1, it could be used to supply eight nozzles of the size for which valve 1 is designed. This scheme — a single flow divider valve with several nozzles — is commonly used in turbojets.

Seat thickness is also important because small particles of contaminant can accumulate under the poppet seat. This will prevent the poppet from seating and cause hysteresis in the flow schedule of the valve, which in turn will distort the fuel distribution in the burner can of a turbojet.

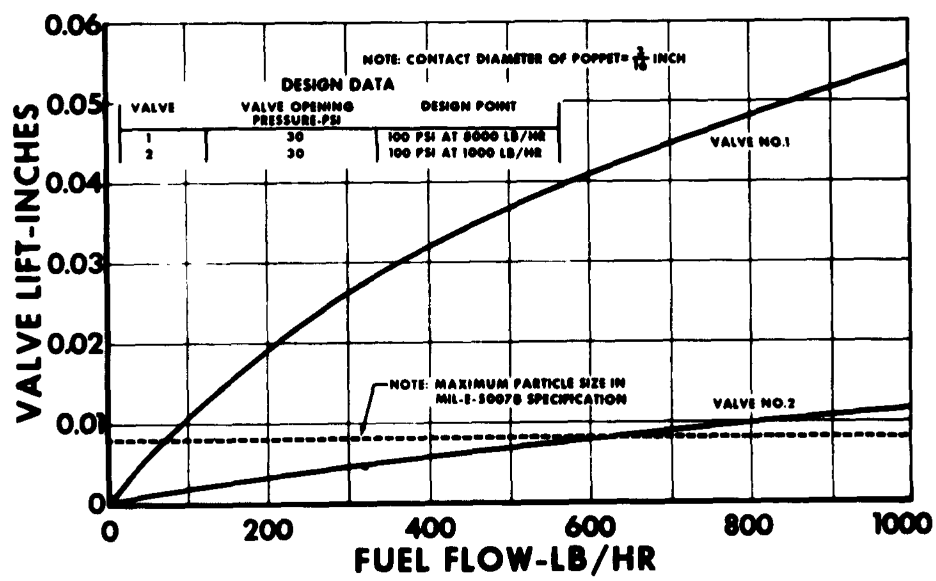


Figure 10 Comparison of Valve Seat Lift for Two Flow Divider Valves **FD 3536**

4. VARIABLE AREA NOZZLE

A variable area nozzle is shown in figure 11. Atomization is achieved in this nozzle by the eruption of a thin fuel sheet that discharges through a small annular opening at the nozzle outlet, rather than by use of centrifugal forces as in swirl type nozzles.

The performance of this nozzle with contaminant will be similar to that predicted for the flow divider valve. Particles of contaminant sticking under the seat of the poppet will cause hysteresis. But more important in the variable area nozzle, contaminant accumulation behind the poppet will cause severe streaks that will not be disturbed before the fuel reaches the burner can.

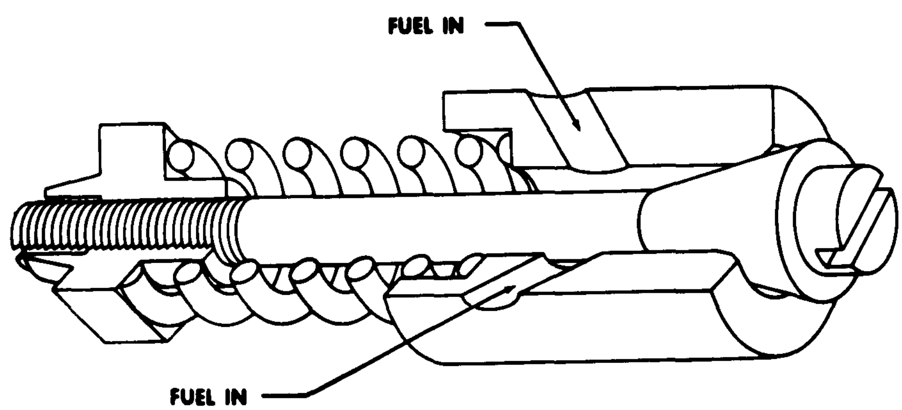


Figure 11 Variable Area Nozzle Schematic FD 3544

SECTION IV

GENERAL STUDY OF THE EFFECTS OF CONTAMINATED FUEL ON TURBOJET PERFORMANCE

A. GENERAL

The effects of contaminated fuel on turbojet performance were evaluated in this study. Consideration was given to the resulting effects of the poor fuel nozzle performance, which was described in Section III, on burner can performance.

B. CONCLUSIONS

Based on the considerations discussed later in this section, the following conclusions can be made:

1. A reduction in spray uniformity can cause (1) a reduction in combustion efficiency and therefore an increase in thrust specific fuel consumption (TSFC); (2) a distorted exit burner can temperature profile, which in turn could damage the turbine inlet guide vanes and the turbine blades or reduce the allowable burner can temperature rise; (3) hot spots on the burner can wall and, ultimately, damage to the burner can; and (4) a reduction in the allowable fuel-to-air operating range.
2. A change in the flow schedule of a fuel nozzle can increase fuel-to-air mixture in a portion of the can and result in damage to the turbine vanes or blades.

C. DISCUSSION

The analytical studies of the effect of contaminated fuel on nozzle performance indicated that contaminated fuel can cause (1) a reduction in spray uniformity caused by streaks, skewed spray pattern, and a reduction in spray angle; and (2) a change in the flow schedule of the nozzle.

The effects of these poor nozzle performance characteristics are presented in subsequent paragraphs.

1. *EFFECT OF A REDUCTION IN SPRAY UNIFORMITY*

a. Combustion Efficiency — The completeness of the combustion process in the burner can depends on the speed at which the process of evaporation, mixing, and chemical reaction takes place.

Evaporation is the most vulnerable to the effects of contamination, because it is significantly affected by the spray pattern. J. S. Clarke[2] has presented the schematic shown in figure 12. This schematic shows the effect of the vortex inside the burner can on droplets of different sizes. Note that the paths of the small droplets are affected significantly by the vortex. The figure illustrates that small fuel droplets have a long residence time in the vortex zone, while large particles have enough momentum to be unaffected by the vortex and, therefore, they traverse the vortex zone in a shorter time than the small droplets. The curve shown in figure 13 was also presented by Clarke. The curve shows that droplets with a diameter of 80 microns and less are completely evaporated, while those of several hundred microns are not affected at all. Consequently, if streaks occur in the spray pattern as a result of the use of contaminated fuel, a larger percentage of the fuel will be unevaporated as it leaves the vortex zone. This will result in incomplete combustion and will be reflected in a reduction in combustion efficiency, and an increase in the TSFC.

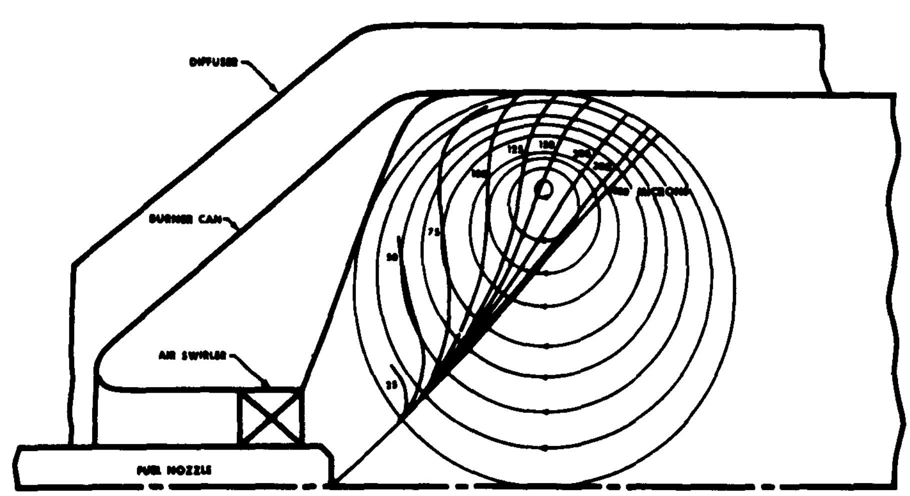


Figure 12 Paths of Various Size Fuel Droplets through the Vortex in the Burner Can FD 3549

[2] J. S. Clarke, "Gas Turbine Combustion Problems," Journal of the Royal Aeronautical Society, Vol. 60, April 1956, page 232.

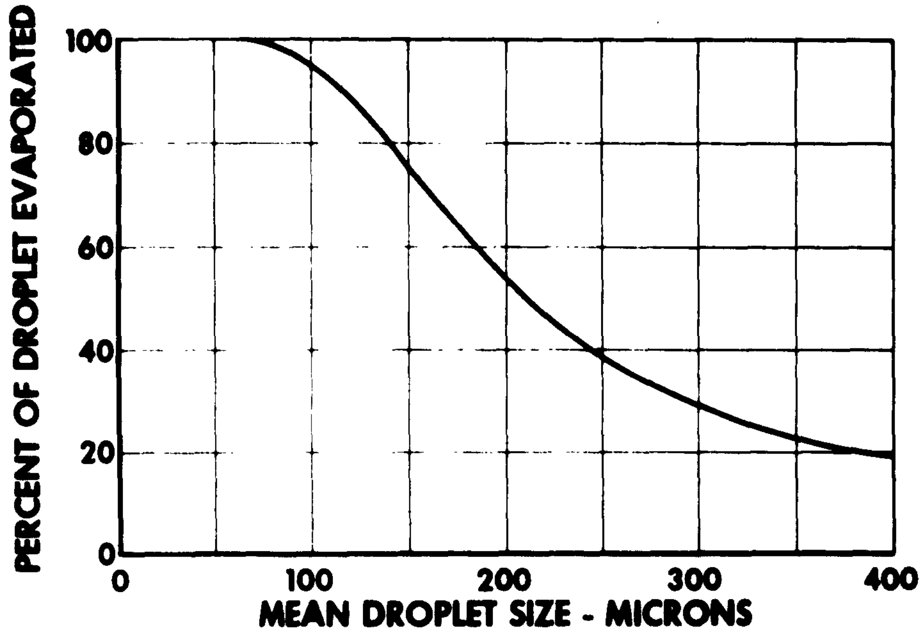


Figure 13 Percent of Droplet Evaporated on Leaving Vortex *FD 3473*

b. Burner Can Exit Temperature Distribution — Burner cans are designed to produce an exit temperature distribution that permits the maximum allowable average turbine inlet temperature and, therefore, maximum engine thrust. This object is attained by adjusting the radial temperature distribution at the burner can exit until it approaches the limits imposed by the turbine inlet guide vanes and the turbine blades. As an example of the temperature limits and their variation across the burner can exit, figure 14 is presented. In this figure the temperature limits imposed by the first two blades and vanes are shown. Also shown in the figure is a practical turbine inlet temperature profile.

Streaks in the spray pattern will move the flame front in the direction of the turbine. This will result in a poor burner can exit temperature distribution. If the streaks are severe enough, the fuel may still be burning as it enters the turbine, and some of the blades and vanes could be melted away. If the streaks were less severe, and if the hot spots could be detected by the thermocouples in the turbine inlet guide vanes, the fuel flow could be reduced, and the damage to the blades and vanes eliminated. This procedure would result in a lower turbine inlet temperature, and naturally, a reduction in engine thrust.

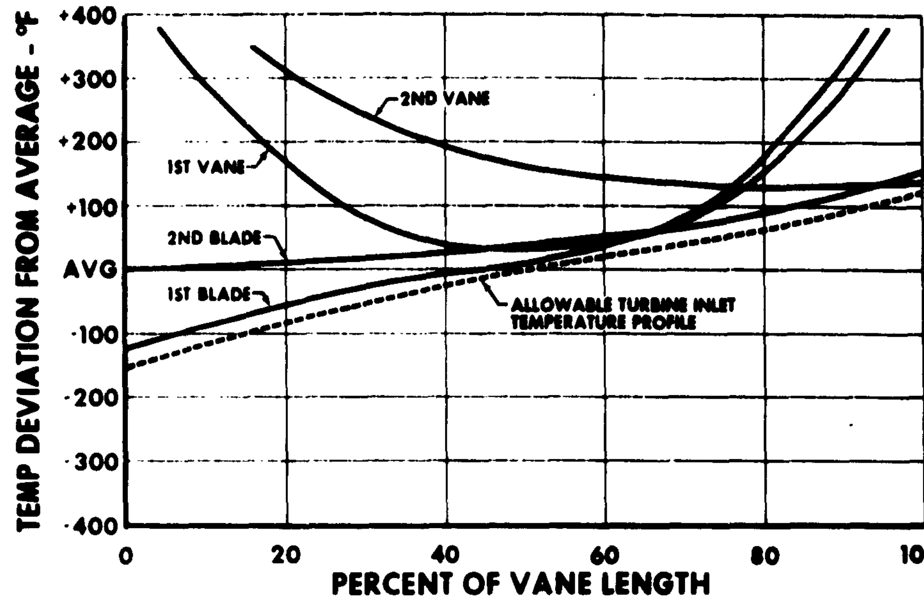


Figure 11 Allowable Temperature Variation at Turbine Inlet *FD 3510*

c. Hot Spots on the Burner Can Wall—A severe streak could cause hot spots in the burner can, which could burn holes in the burner can wall. This would be detrimental, because air entering the burner through this new hole would change the entire airflow pattern in the burner can, which in turn would reduce combustion efficiency, and cause hot spots at the burner can exit.

d. Reduction in the Fuel-to-Air Operating Range -- Poor fuel distribution in the burner can resulting from streaks or a skewed spray pattern could also effect a reduction in the combustor fuel-to-air operational range. The most important of these limits is the lean fuel-to-air limit, since this limit restricts the operational altitude of the turbojet.

2. *EFFECT OF A CHANGE IN FLOW SCHEDULE*

The change in nozzle flow schedule caused by contaminant accumulation in the nozzle will result in (1) an increase in pressure in the fuel system, and (2) a poor fuel distribution in the burner can.

The first of these effects will not be harmful to the engine fuel pump unless the pressure increases to the pump design limit. Most fuel systems control flow by controlling the pressure drop across a fixed area. Normally the fuel pump is driven by the engine; consequently, the pump speed is almost constant regardless of the engine power setting. This results in a constant fuel pump discharge rate; and the fuel that is not being consumed by the engine is recirculated to the pump suction. If contaminant accumulates in a nozzle, the pressure drop across the nozzle will increase and thus cause the fuel pump discharge pressure to increase. If the pressure reaches

the pump design limit, a pressure-actuated relief valve in the pump discharge line will open and fuel will be bypassed to the pump suction. If this occurs, a drop in net thrust will occur.

A change in the nozzle flow schedule will cause poor fuel distribution in the burner because normally a number of fuel nozzles are operated in parallel. That is, several nozzles are supplied by one fuel manifold. The fuel distribution in the combustor is, therefore, dependent on the matching of the flow schedule of the nozzles. With contaminant accumulation, the flow schedule of a nozzle will change; consequently, the fuel distribution will be decreased. As an example of this effect, consider the case of 48 simplex type fuel nozzles being supplied by a control system. If one of the nozzles becomes completely plugged, then the fuel flow through the other 47 nozzles will be increased to 48/47 of the original flow. The fuel control system discharge pressure will be increased to $(48/47)^2$ of the original value. In a similar manner the fuel flow through the unplugged nozzles and the pressure drop across the nozzles can be calculated for the cases when several nozzles are plugged. The results of these calculations are shown in figure 15.

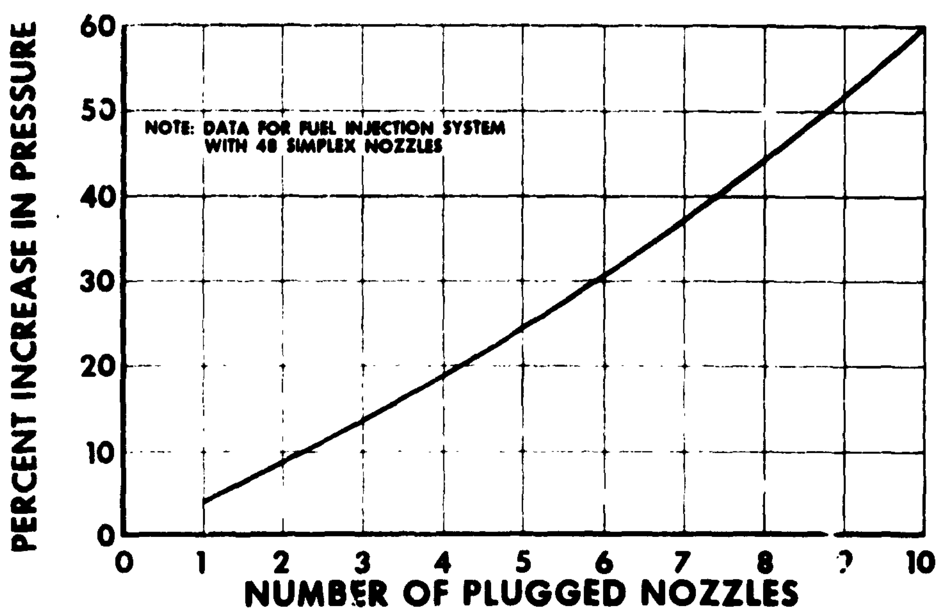


Figure 15 Effect of Plugged Nozzles on Fuel Control System Discharge Pressure *FD 3500*

An increase in fuel flow through the unplugged nozzles will result in a higher fuel-to-air ratio in a portion of the engine and could result in a temperature rise in that portion of the engine that is above the allowable. By the simple calculations illustrated above and by assuming that the burner can temperature rise is directly proportional to the fuel-to-air ratio,

the effect of several plugged nozzles on turbine blade life can be determined. The results of this procedure are illustrated in figure 16. The figure indicates that even if several nozzles are plugged the engine still could be operated without damaging the turbine.

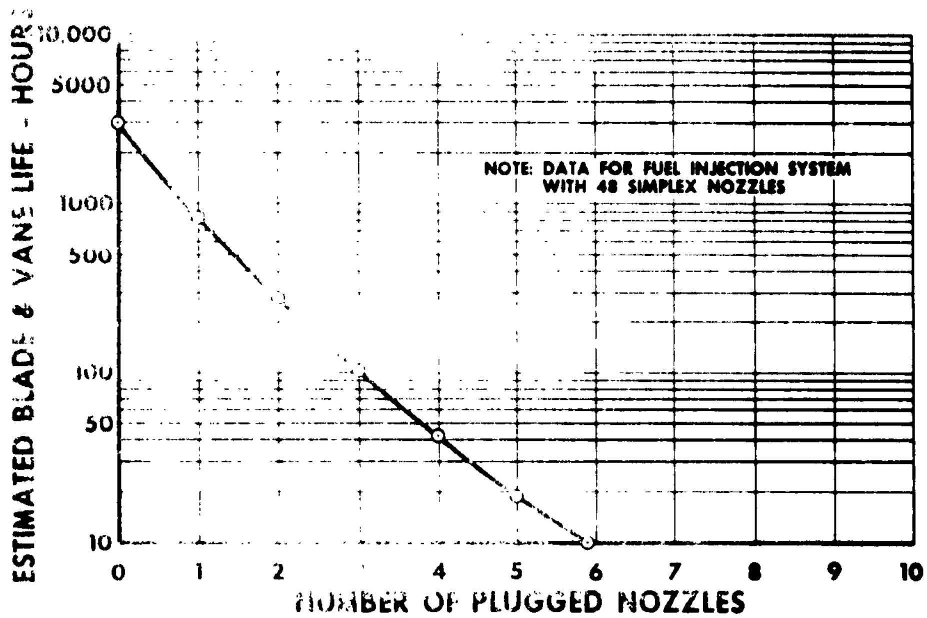


Figure 16. Effect of Plugged Nozzles on Blade and Vane Life **FD 3498**

SECTION V

NOZZLE SPRAY TESTS

A. GENERAL

For fulfillment of Work Statement Item 4 of the contract, nozzle spray tests were made with contaminated fuel. The purpose of these tests was to determine experimentally the effects of contaminated fuel on the performance of fuel nozzles that are used in current turbojet engines.

In these tests, a constant inlet nozzle pressure was maintained; therefore, the contaminant accumulation in the nozzle was indicated by variation of fuel flow. Records were kept of the spray pattern behavior during the tests. Nozzle calibrations were made during the test and at the conclusion of the test. Each nozzle was disassembled and examined at the conclusion of the test to determine where the contaminant accumulated in the nozzle.

Included in this section of the report are (1) the conclusions of the nozzle spray tests, (2) a description of the test stand, (3) the results of the comparative nozzle spray tests, and (4) the results of additional tests.

B. CONCLUSIONS

1. *NOZZLE TYPES*

a. In swirl type nozzles the swirl slots are the first location where contaminant accumulates. This can cause a significant decrease in flow rate (if a constant inlet pressure is maintained); however, it will not affect the spray pattern significantly. Swirl slots as large as 0.018 x 0.018 in. were affected by contaminants. The size of the slot that would perform for 60 hours with contaminated fuel depends on nozzle shape; circular slots smaller than 0.018 in. would probably give satisfactory performance. If a 350-micron filter is used at the nozzle inlet, swirl slots that are at least 0.018 in. and probably as small as 0.014 in. (the screen opening size) could be used.

b. In dual orifice nozzles, contaminant can collect at the annulus between the primary and secondary orifice. Accumulation of contaminant at this nozzle location can cause pronounced effects on the nozzle spray pattern; streaks and gaps in the spray pattern will appear. It was found that an annulus of 0.027 in. is large enough to permit satisfactory operation for 60 hours.

c. The performance of the variable area nozzle was completely unsatisfactory with contaminated fuel. This performance is considered to be typical of what could be expected with all nozzles in which the outlet area is variable. A nozzle in which the slot area

[illegible] (such as the variable area dual orifice that was evaluated) does not fit into this category.

d. Swirl type nozzles that are properly designed are capable of spraying contaminated fuel; furthermore, the spill nozzle is probably the best swirl nozzle type for most flow ranges.

e. Accumulation of contaminant at the seat of the flow divider valve can cause hysteresis in the flow schedule.

f. The distance between moving parts must be less than 0.00018 in. or greater than 0.0128 in. These clearances are adequate for contaminated fuel that has been filtered through a 150-micron central filter and then through a 350-micron nozzle filter.

2. *FILTER SIZES*

Nozzle filter design is important; if the filter surface area or the filter openings are too small, the filter will become plugged and will rupture. Thus, the contaminant that has been collected on the filter can be carried to the fuel nozzle. If this occurs, nozzle performance will deteriorate rapidly. A 350-micron filter with a surface-area-to-fuel-flow ratio of 0.00365 $in^2/lb/hr$ is adequate for 60 hours of operation with contaminated fuel that has been filtered through a 150-micron central filter. A 710-micron filter is too large.

3. *CONTAMINANT EFFECTS*

In the spray test several observations were made regarding the contaminants and their effects. These are discussed below:

a. Unfiltered MIL-F-5007B contaminant plus the fungus solution is too severe for all of the turbojet fuel nozzles tested.

b. Fungus causes pronounced effects on nozzle performance. During the cluster conditioning test, a set of variable area dual orifice nozzles performed well with MIL-F-5007B filtered through a 150-micron screen. The addition of the fungus caused abrupt fluctuation in flow and pressure.

c. Lint and fungus together can bridge large openings; even after being filtered through a 150-micron screen.

d. The salt water corrodes away nozzle parts. Corrosion was detected on martensitic stainless steels; austenitic type stainless steels are considered to be a better choice for this service.

e. The presence of the contaminants probably increases the erosion of nozzle parts.

C. DISCUSSION

1. DESCRIPTION OF TEST EQUIPMENT

A piping diagram of the test stand used during the contaminated fuel spray tests is shown in figure 17. As shown in the figure, a circulating fuel system was used. The contaminated fuel from the test piece was returned to the main fuel supply tank; it was then pumped through a 2-micron filter where all of the solid contaminants (dust, sand, etc.) were removed. The contaminant addition system, which is shown boxed in with dotted lines in the figure, was in parallel with the main fuel supply pump. This system included a variable speed conveyor that dropped the solid contaminants into a tank where they were mixed with fuel. Some of the fuel from the contaminated fuel pump was returned to the tank to provide agitation. The remainder of the fuel flowed through the 150-micron filter and to the test piece. It should be noted that the fuel was contaminated to the level specified in MIL-E-5007B by adding contaminants in proportion to the total flow being supplied to the main supply pump and the contaminant addition system.

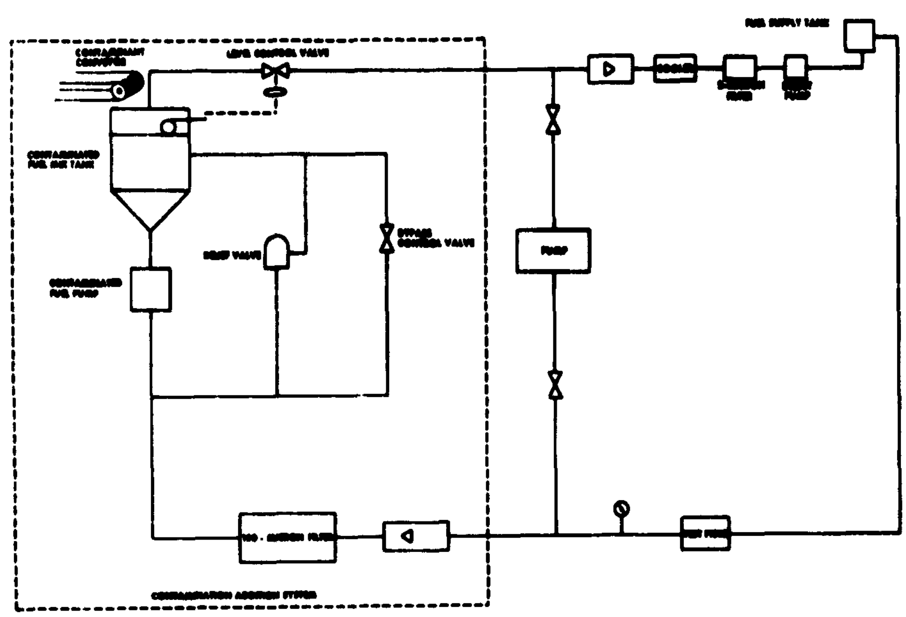


Figure 17 Contaminated Fuel Test Stand *FD 3509*

The conveyor belt was used to add dust, sand, ferric oxide, and cotton linters. The fungus and salt water solution was supplied by pouring measured quantities that were proportional to the total flow into the contamination mixing tank at 5-minute intervals. The naphthenic acid, which is a liquid contaminant, was added to the total fuel in the test stand only at the beginning of the test, since the 2-micron filter would not remove it.

For the tests, a manifold was built for supplying contaminated fuel of the same concentration to six different nozzles. This manifold is shown schematically in figure 18. The contaminated fuel enters this manifold at the bottom (the manifold's axis of revolution was vertical during the test). Each nozzle was fed from one of the six lines that are attached to the manifold at the same vertical level and are equally spaced around the outer periphery of the manifold.

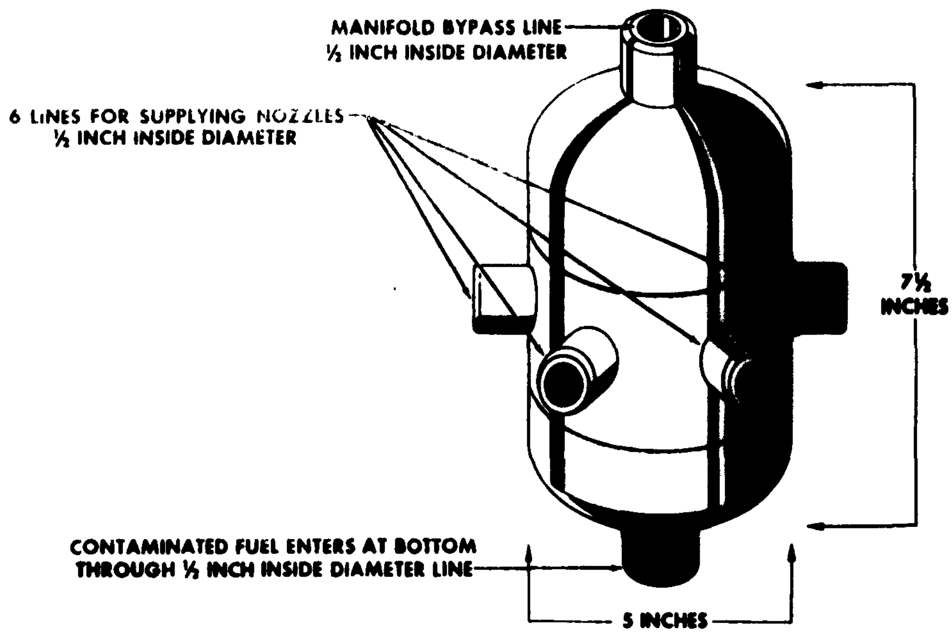


Figure 18 Fuel Manifold FD 3025

The piping arrangement that was used during the test is shown in figure 19. Since each of the nozzle inlet pressures was different, a different pressure drop was maintained across each nozzle pressurizing valve (the valve between the manifold and each nozzle). This condition was achieved with a wide setting of the pressurizing valve by maintaining a large flow rate through each nozzle bypass valve. After the test was started, the setting of the pressurizing valves was not disturbed; the inlet pressure to each nozzle was maintained by regulating the individual nozzle bypass valves.

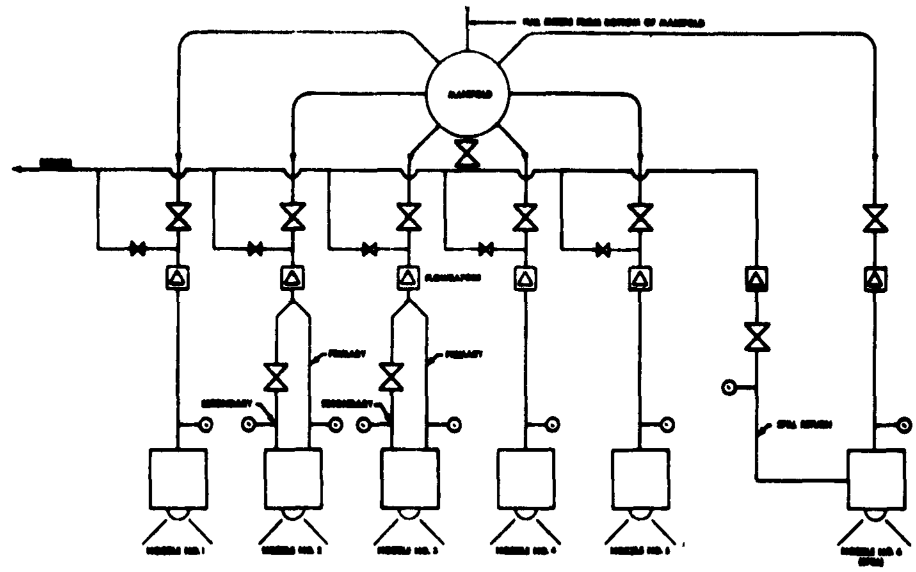


Figure 19 Comparative Nozzle Test Flow Diagram *FD 3534*

During the test, fuel samples were taken at several locations in the system and contaminant concentration was determined by millipore tests. The results of the millipore tests are shown in figure 20 where the concentrations are plotted against time during the comparative nozzle spray tests. The samples were taken at the discharge of the spill nozzle. The 150-micron filter elements were changed at the times indicated. (The presence of this filter is the reason for the lower than specification MIL-E-5007B contaminant concentration.) The high concentration obtained during the first 10 hours is attributed to a buildup and then release of the accumulated contaminants through the system. No attempts were made to improve the uniformity

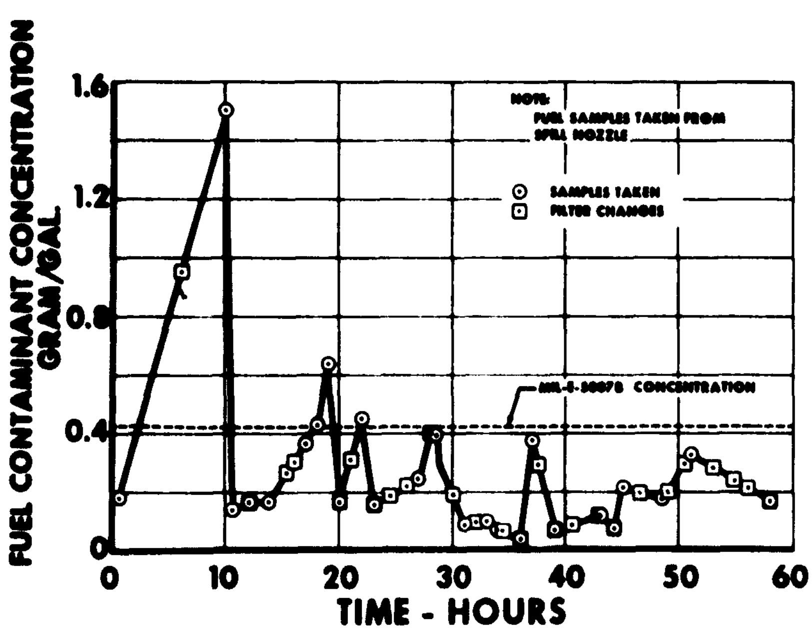


Figure 20 Variation in Fuel Contaminant Concentration *FD 3511*

of the contaminant concentration level, because in an engine the contaminant would accumulate at several locations in the fuel system and then be carried through to the nozzles.

The nozzles were mounted in a spray booth so that spray patterns could be observed.

2. *COMPARATIVE NOZZLE SPRAY TESTS*

A 60-hour test, which consisted of flowing contaminated fuel through six different types of nozzles, was made. In this test, the contaminated fuel was filtered through a 150-micron filter in the main fuel line. At the conclusion of the 60-hour test, the 150-micron filter was removed from the rig and an accelerated test was made on the same six nozzles. The accelerated test lasted for 17 hours.

The nozzles that were tested are described in table I.

TABLE I. NOZZLE DESCRIPTIONS

Nozzle No.	Nozzle Type	Approximate Flow Range, lb/hr	Screen Size
1	Dual Orifice with Integral Flow Divider	50 to 1050	710 micron
2	Dual Orifice	70 to 535	—
3	Dual Orifice	65 to 1110	370 micron
4	Dual Orifice Variable Area	35 to 875	—
5	Variable Area	35 to 875	—
6	Spill	0 to 730	—

NOTE

Nozzles 1, 5, and 6 supplied by the Parker-Hannifin Corporation; nozzles 2, 3, and 4 were supplied by the Ex-Cell-O Corporation. As denoted, only nozzles 1 and 3 contained filters. Since these filters are much larger than the 100-mesh (154-micron) filter that was installed at the pump discharge, these filters were left in the nozzles. These filters are the size that is normally supplied on these two production nozzles.

The passage sizes for all of these nozzles are not available because the nozzles were obtained on consignment; however, the relative passage size can be obtained by a comparison of the nozzle parameter derived from dividing the flow rate by the square root of the nozzle pressure drop for the flow rate selected. (This parameter is equal to the product of the discharge coefficient and the outlet orifice area of a nozzle; it will be constant for a fixed area orifice.) Table II presents this parameter for each of the nozzles, in addition to the flow rate, and pressure at which each parameter was obtained.

TABLE II. RELATIVE NOZZLE SIZE

Nozzle No.	Orifice	Fuel Flow, lb/hr	Pressure drop at the Fuel Flow Indicated, psi	Nozzle Size Parameter, $in^2 \times 10^{-4}$
1	Primary	90	375	2.8
	Secondary	160	375	5.9
2	Primary	67	100	4.0
	Secondary	205	100	12.3
3	Primary	130	100	7.9
	Secondary	510	100	31.0
4	Primary	43	240	1.7
	Secondary	207	240	8.0
5	–	250	240	9.6
6	–	730	648	17.0

During the test, the inlet pressure to each nozzle was maintained constant. The contaminant accumulation was indicated by a change in the flow rate. The inlet pressure that was maintained and the corresponding flow rate for each nozzle are given in table III.

TABLE III. TEST CONDITIONS

Nozzle No.	Inlet Pressure, psig	Flow Rate, lb/hr
1	375	250
2	257	110 (Primary) 150 (Both)
3	174	170 (Primary) 330 (Both)
4	240	250
5	265	250
6	385 200 (Spill Chamb. Press.)	730 (Inlet Flow) 150 (Outlet Flow)

The performance of each of the nozzles during the tests, and the results of the post-test examinations are presented below:

a. Nozzle No. 1 (Dual Orifice Nozzle with Integral Flow Divider)

A schematic of this nozzle is shown in figure 21. The flow divider valve is approximately three inches upstream of the dual orifice. A 710-micron screen filters the fuel flow to the secondary orifice; and, according to Parker Hannifin, the primary orifice is protected by a coarse sieve arrangement.

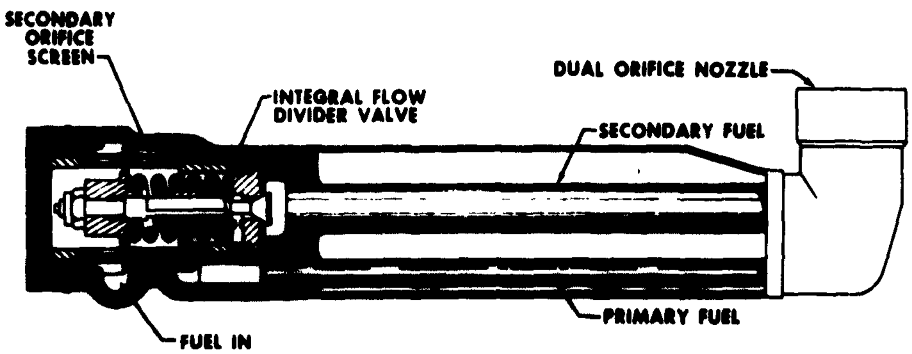


Figure 21 Dual Orifice Nozzle with Integral Flow Divider Valve (Nozzle No. 1) *FD 3183*

A history of the fuel flow from this nozzle is shown in figure 22. As shown in the figure, the flow dropped abruptly during the first two hours of the test. The flow dropped to approximately the flow rate of the primary orifice, indicating that either the flow divider valve or the secondary orifice was completely plugged. As shown in the figure, the primary orifice continued to flow until the accelerated test was started; then it began to plug.

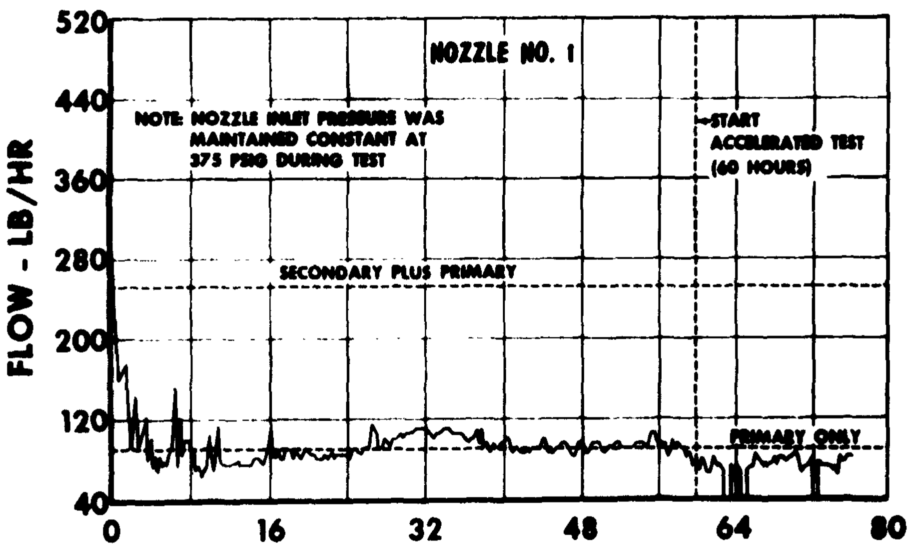


Figure 22 Fuel Flow Variation, Dual Orifice Nozzle with Integral Flow Divider (Nozzle No. 1) *FD 3537*

During the first 60 hours, this nozzle consistently produced a fine spray with a light streak. Periodically, there would be gaps in the spray cone. During the accelerated test, this nozzle performed poorly. The flow rate was only 25 percent of normal at an inlet pressure of 375 psig, and the spray pattern was skewed. The final calibration data are compared with (1) the data obtained after the 60-hour test, and (2) the initial calibration data in figure 23. The large decrease in flow indicates how severely the nozzle was plugged.

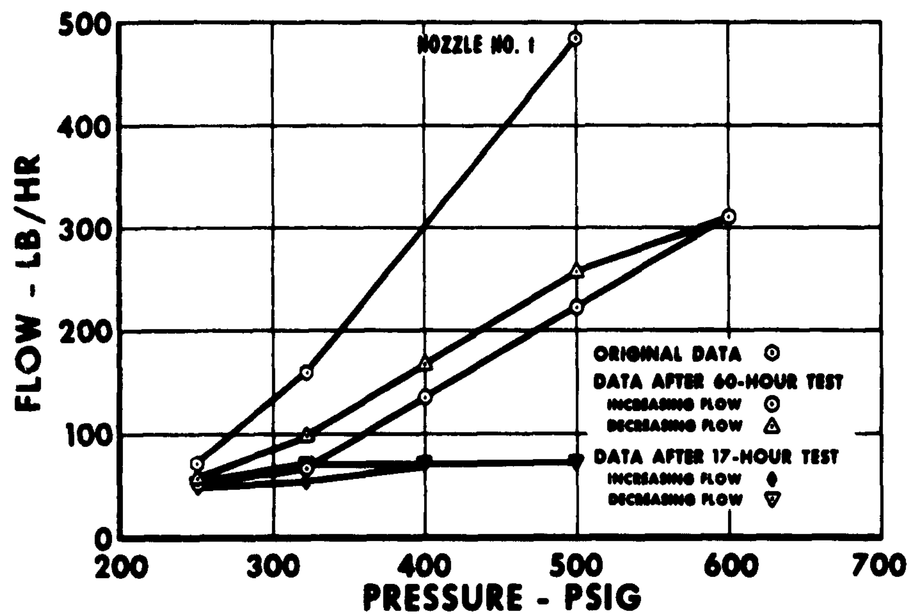


Figure 23 Calibration Data, Dual Orifice Nozzle with Flow Divider (Nozzle No. 1) *FD 3030*

Figure 24 shows the nozzle after it was disassembled following the accelerated test. The dual orifice is contained in the part indicated in figure 24. An enlarged view of this part looking downstream is shown in figure 25. (An attempt was made to remove the metering parts from the housing but it was found that this would result in damage to the nozzle. Therefore, since the nozzle is on consignment from the Parker-Hannifin Corporation, further disassembly was not made.) Close examination revealed that contamination had almost bridged the inlet slots that lead to the swirler of the secondary orifice, and also the annulus between the primary and secondary orifices. Consequently, the flow from the secondary orifice was low, which explains the large decrease in fuel flow that occurred during the test. The presence of the 710-micron filter apparently was not effective in keeping the secondary orifice clean. The primary orifice inlet and outlet were clean.

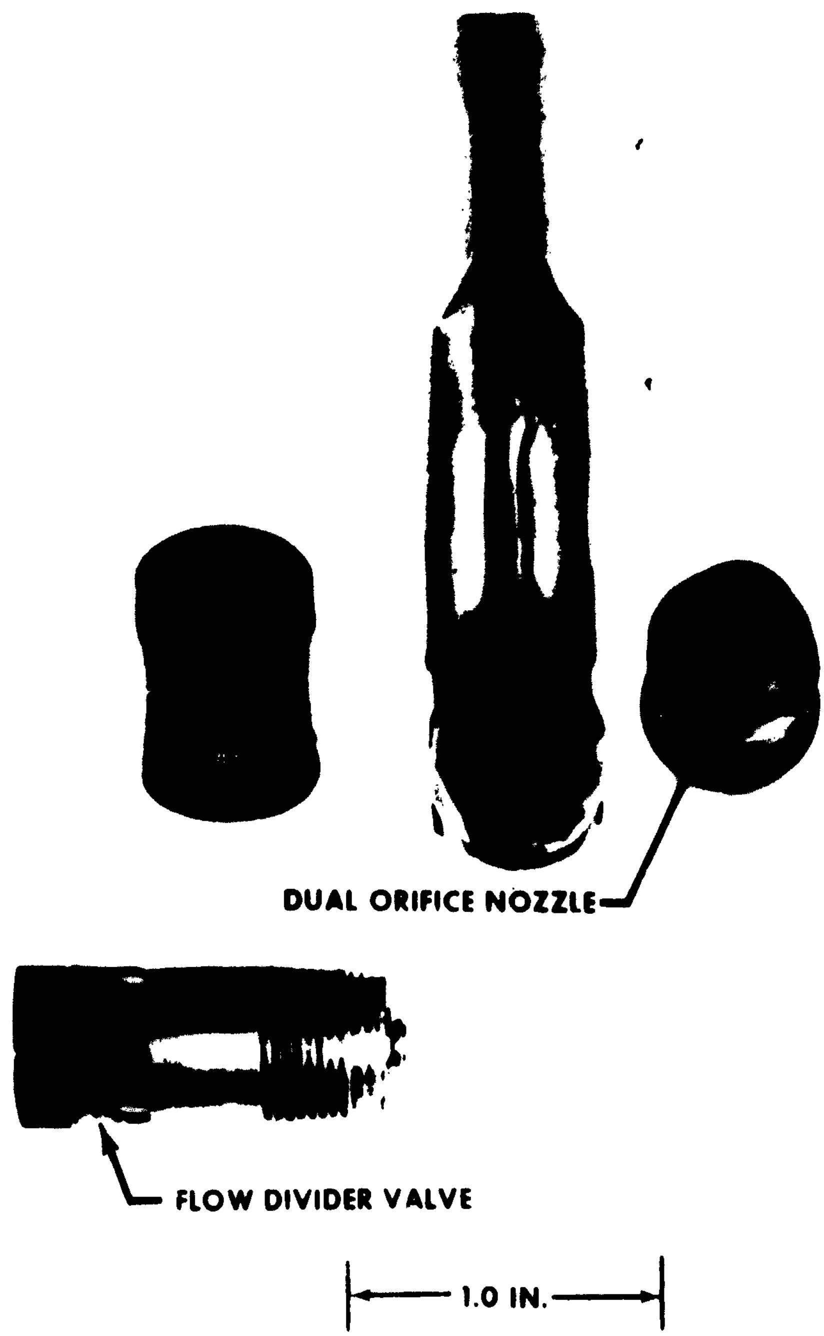


Figure 24 Dual Orifice Nozzle with Integral Flow Divider (Nozzle No. 1)—Disassembled after Test *FD 3180*

Figure 25 Enlarged View of Dual Orifice Nozzle (Nozzle No. 1) **FD 3162**

The flow divider valve was relatively clean. A small ring of ferric oxide adhered to the poppet at the seat location. This would have resulted in hysteresis if the secondary nozzle had not been so severely plugged. This ring of contamination may be the reason for the hysteresis observed during the calibration following the 60-hour test.

Although the secondary orifice is larger than the primary orifice, it plugged during the test; the primary orifice continued to flow. This is attributed to plugging of the small annulus between the two orifices. If the primary orifice had been inside the secondary, the secondary orifice would not have plugged.

b. Nozzle No. 2 (Dual Orifice Nozzle, 70 to 535 lb/hr Flow Range)

This nozzle is typical of dual orifice nozzles that are used on turbojet engines. A cutaway drawing of the nozzle is shown in figure 26. The important dimensions of this nozzle are given in table IV.

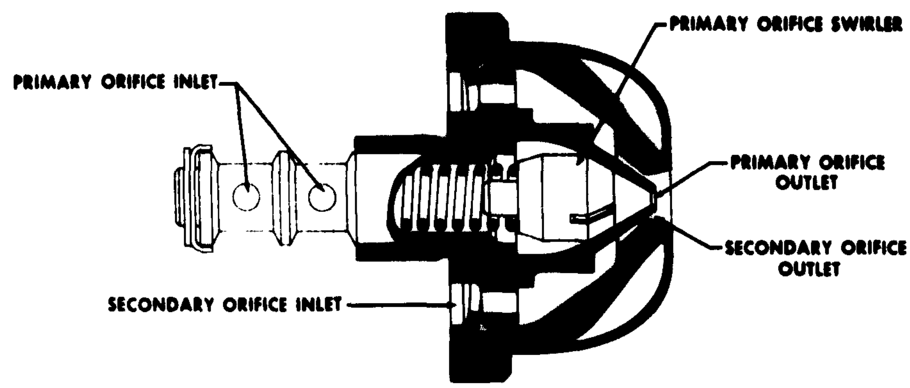


Figure 26 Dual Orifice Nozzle Cutaway (Nozzle No. 2) *FD 3169*

TABLE IV. NOZZLE SPECIFICATION

	Primary	Secondary
Outlet orifice diameter, inch	0.048	0.105
No. of swirl slots	3	6
Area of a swirl slot, in^2	0.000126	0.000746
Diameter of swirl chamber, inch	0.125	0.250
Dimension of a square slot with the same area, inch	0.0123	0.0274

The thickness of the annulus between the primary and secondary is 0.0065 inch. The primary slots are almost triangular in cross section; the secondary swirl slots are square.

A hand-operated metering valve was put in the supply line leading to the secondary orifice to simulate a flow divider valve; during the test contaminant accumulated in this valve. Constant monitoring was required to keep this valve clear of contaminant and keep fuel flowing to the secondary orifice. Consequently it was decided to close the metering valve and test the primary orifice as a simplex nozzle. This decision produced equally valuable program results because both the secondary and primary orifices are the same basic type. The only additional location for contamination to accumulate in the secondary orifice is in the annulus between the primary and secondary orifices.

A plot of nozzle fuel flow during the test for a constant inlet nozzle pressure of 257 psig is presented in figure 27. As shown in the figure, after the valve in the secondary supply line was closed, the primary flow was normal until the valve was opened during the 26th hour of the test. The valve was opened to determine if contamination would accumulate in the annulus between the primary and secondary orifices. Streaks occurred frequently when this valve was open; indicating that contaminant was being held up momentarily at the annulus. After the 36th hour (with the valve in the secondary orifice supply line closed), the primary began to clog and the fuel flow dropped.

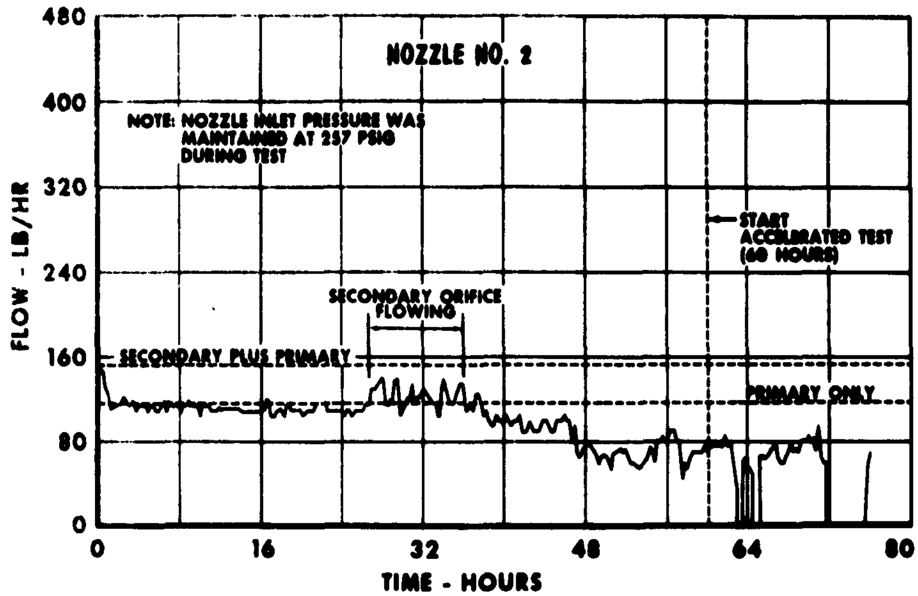


Figure 27 Fuel Flow Variation, Dual Orifice Nozzle (Nozzle No. 2) **FD 3538**

During the accelerated test the primary stopped flowing completely three times as shown; one time the flow remained zero for 3¾ hours. During these times a constant inlet nozzle pressure of 257 psig was maintained. Apparently some large piece of contaminant, possibly fungus, had plugged the outlet orifice or the swirl slots.

The results of the post-test calibrations are presented in figure 28. It should be noted that there is little difference between the calibration data taken after the 60-hour test and that obtained after the 17-hour (accelerated) test. Apparently the contamination that caused complete flow stoppage was carried through the nozzle.

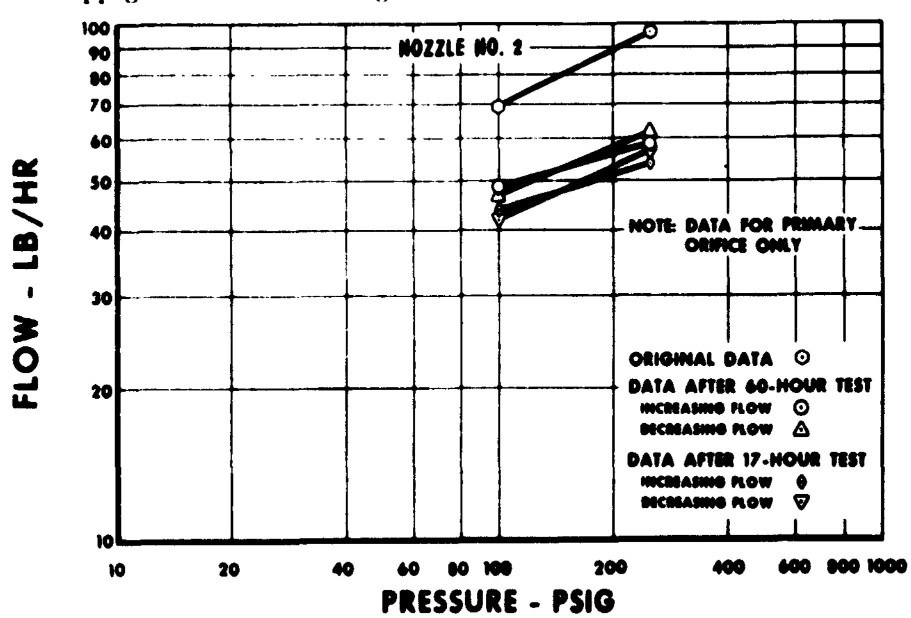


Figure 28 Calibration Data, Dual Orifice Nozzle (Nozzle No. 2) **FD 3027**

The nozzle is shown disassembled in figure 29. As shown, contamination accumulated in the swirler slots, although not severely. It should be noted that the large decrease in flow at a constant inlet pressure accompanying the small accumulation of contaminants in the swirl slots indicates the sensitivity of nozzles to plugged passages

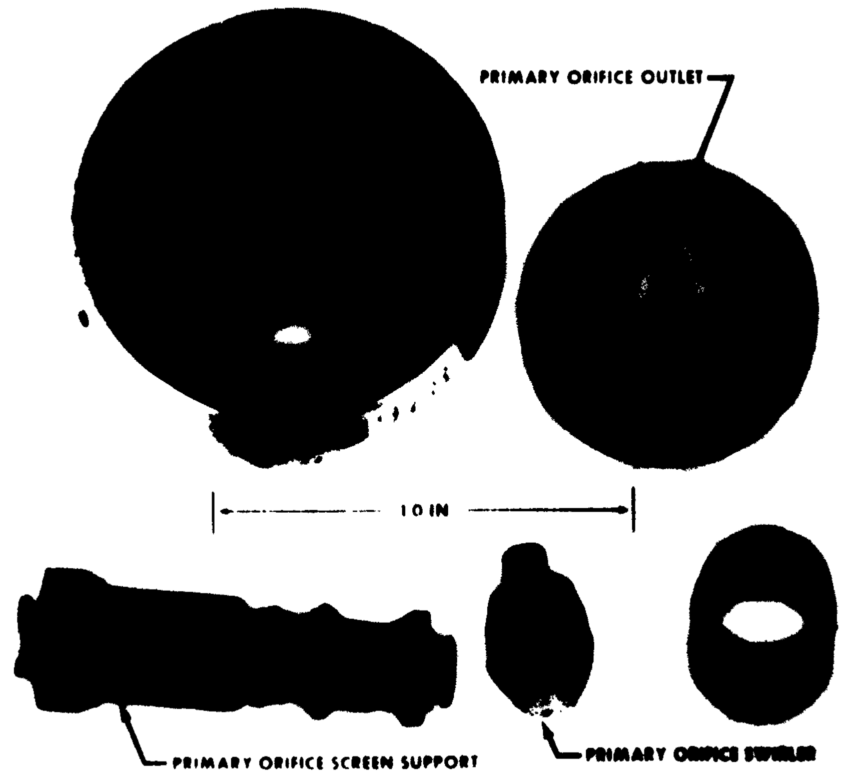


Figure 29 Dual Orifice Nozzle (Nozzle No. 2)—Disassembled after Test *FE 18353*

Other than these two locations, the nozzle metering parts were fairly clean. Complete plugging occurred in one of the holes in the primary nozzle screen support, through which fuel flows before entering the primary nozzle. This did not affect nozzle performance; probably because there are several additional holes for fuel passage. Furthermore, since the holes are fairly large (approximately 0.10 inch diameter), an increase in pressure drop would probably force the contaminant through the hole.

c. Nozzle No. 3 (Dual Orifice Nozzle, 67 to 1110 lb/hr Flow Range)

This nozzle is a large version of Nozzle No. 2. The schematic shown in figure 26 shows how the nozzle is made.

The dimensions of this nozzle are given in table V.

TABLE V. SPECIFICATIONS FOR NOZZLE NO. 3

	Primary	Secondary
Outlet orifice diameter, inch	0.0543	0.135
No. of swirl slots	4	8
Area of a swirl slot, in^2	0.000355	0.001715
Diameter of swirl chamber, inch	0.1565	0.344
Dimension of a rectangular slot with the same area, inch	0.0189	0.0415

The thickness of the annulus between the primary and secondary orifices is 0.0186 inch. The primary swirl slots have a triangular cross-sectional area; the secondary swirl slots are square. A 370-micron screen that filters the fuel to both orifices was left in the nozzle during the test.

A valve was placed in the line supplying the secondary orifice (as was done with Nozzle No. 2). This valve also required close monitoring so only the primary orifice was tested during most of the test.

Figure 30 is a record of the fuel flow to nozzle No. 3 during the test. The nozzle performed well during the first 60 hours of the test. Fuel flow dropped toward the end of this period. No streaks were detected in the spray pattern at any time during the first 60 hours, even when the valve in the secondary supply line was open. The 370-micron screens apparently were effective in keeping the large particles from entering the secondary orifice and being held up in the orifice annulus.

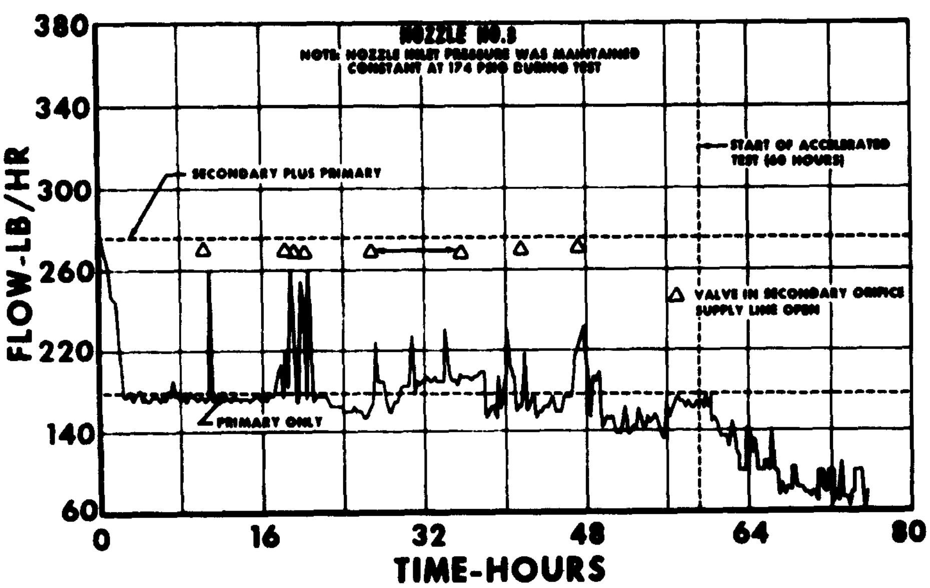


Figure 30 Fuel Flow Variation, Dual Orifice Nozzle (Nozzle No. 3) *FD 3545*

The performance of this nozzle during the 17-hour test was poor. With the inlet pressure maintained constant at 174 psig, the fuel flow from the nozzle gradually decreased from approximately 150 to 50 lb/hr. The severity of the contaminant accumulation is indicated by the final calibration data shown in figure 31.

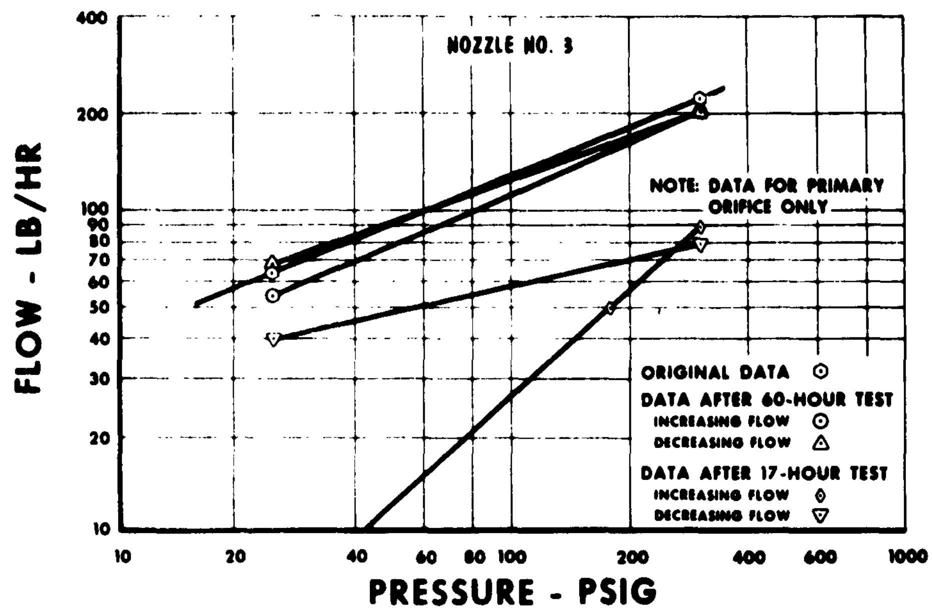


Figure 31 Calibration Data, Dual Orifice Nozzle (Nozzle No. 3) ***FD 3028***

The nozzle is shown disassembled in figure 32. As shown, there were heavy contaminant deposits on the primary orifice screen, the primary orifice screen support, and on the spring that holds the primary orifice swirler in position. Also one slot of the primary orifice swirler was completely filled with contamination, and heavy deposits occurred around the perimeter of the primary orifice swirler. Even though the screen was found to be ruptured upon disassembly, it is believed that it remained intact during the 60-hour test and was effective in protecting the metering parts of the primary orifice. This theory is supported by (1) the fact that the performance of this nozzle during the 60-hour test was good, and (2) the good agreement between the initial calibration data and the calibration data obtained after the 60-hour test. The screen was ruptured during the accelerated test and the contaminant that had adhered to the outside of the screen was carried to the orifice metering parts. This in turn caused the severe plugging that is evidenced by the final calibration data.

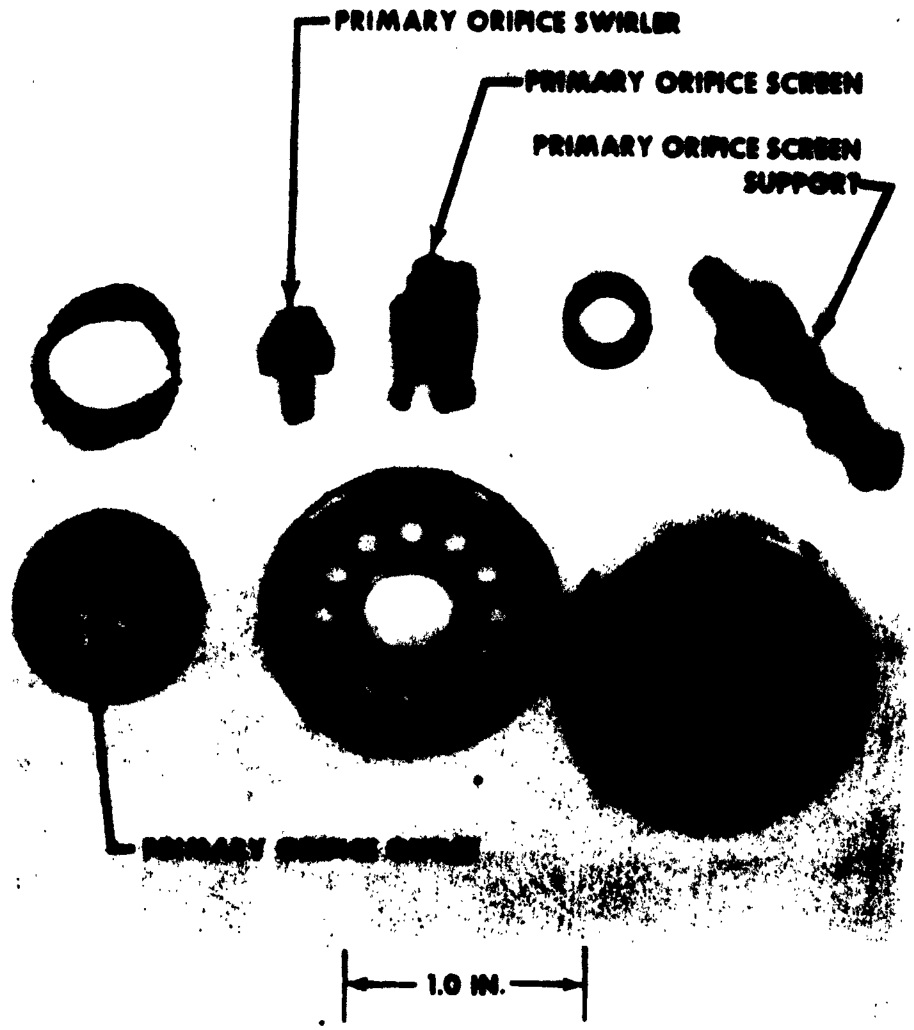


Figure 32 Dual Orifice Nozzle (Nozzle No. 3)—Disassembled After Test ***FE 18356***

d. Nozzle No. 4 (Dual Orifice, Variable Area Nozzle)

This is a dual orifice nozzle in which the secondary swirl slot area varies with the nozzle pressure drop. A schematic of this nozzle is shown in figure 33. Part A in the figure contains the primary orifice, this part is also spring located and schedules flow to the secondary orifice. At pressures less than 150 psi, this part is seated against part B at the location X. As the pressure is increased, part B moves outward away from its seat; this movement uncovers the secondary slots that are located as indicated in the figure. The important dimensions of this nozzle are listed in Table VI.

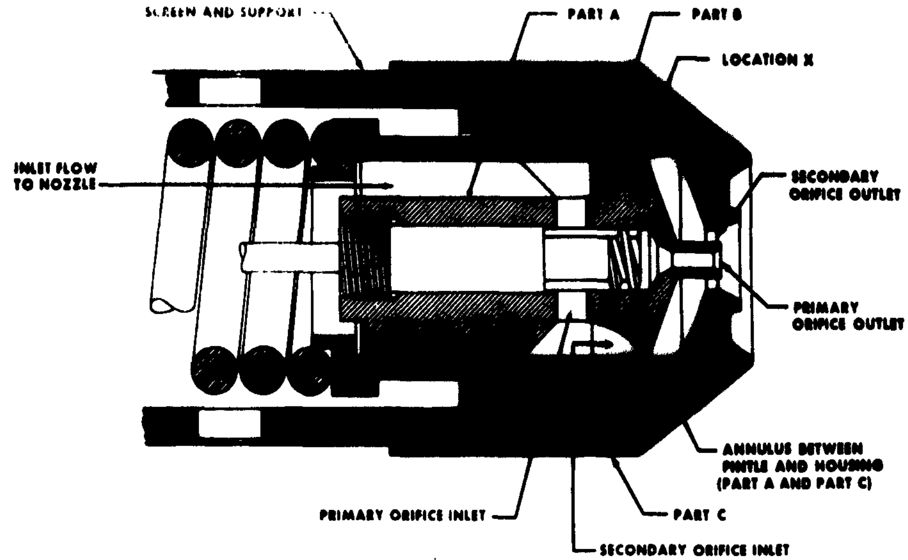


Figure 33 Variable Area Dual Orifice Nozzle Schematic (Nozzle No. 4) ***FD 3561***

TABLE VI. SPECIFICATIONS OF NOZZLE NO. 4

	Primary	Secondary
Outlet orifice diameter, inch	0.035	0.120
No. of swirl slots	2	6
Area of a swirl slot, in^2	0.000328	0 to 0.010 (Variable)
Diameter of a swirl chamber, inch	0.125	0.400
Dimensions of a square slot with the same area, inch	0.0182	0 to 0.1

The thickness between the primary and the secondary orifices is 0.0266 inch. The primary orifice swirl slots are square; the secondary slots are almost triangular in cross-section.

NOTE

The annulus between the pintle and the secondary swirl chamber diameter that is referred to repeatedly in this report is the annulus between the parts A and C as indicated in figure 33. Initially this annulus was 0.005 inch (125 microns) thick.

During the test, the flow was either excessive or lower than normal as shown on figure 34. This performance is attributed to the position of the pintle. Excessive flow occurred when the pintle was stuck open, while flow below normal occurred when the pintle was in a normal position, and contamination that had accumulated at another nozzle location limited the flow. In addition, several times it was necessary to clear the nozzle by reducing and then increasing flow, because of accumulation of contaminants between the pintle and the secondary orifice swirl chamber. This nozzle location is indicated in figure 33. The clearance at that point is only 125 microns. Particles caught there and held the pintle open. When the pressure was reduced, the spring forced the pintle backward toward its seat. This action cleaned the contaminant from the nozzle.

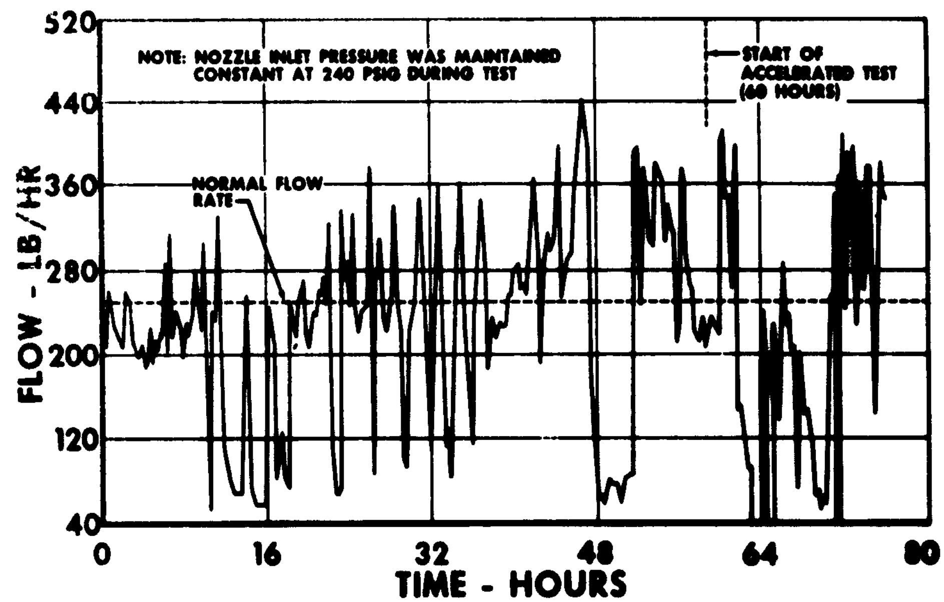


Figure 34 Fuel Flow Variation, Variable Area Dual Orifice Nozzle (Nozzle No. 4) FD 3524

The calibration data for this nozzle are given in figure 35. At the beginning of the calibration following the 60-hour test, the pressure was increased from zero to 200 psig. At this pressure the pintle became stuck open, permitting the flow to gush out. When the pressure was reduced, the nozzle cleared and the calibration was completed. In the calibration after the accelerated test the primary orifice was apparently plugged because no flow was obtained until the pressure reached 200 psig.

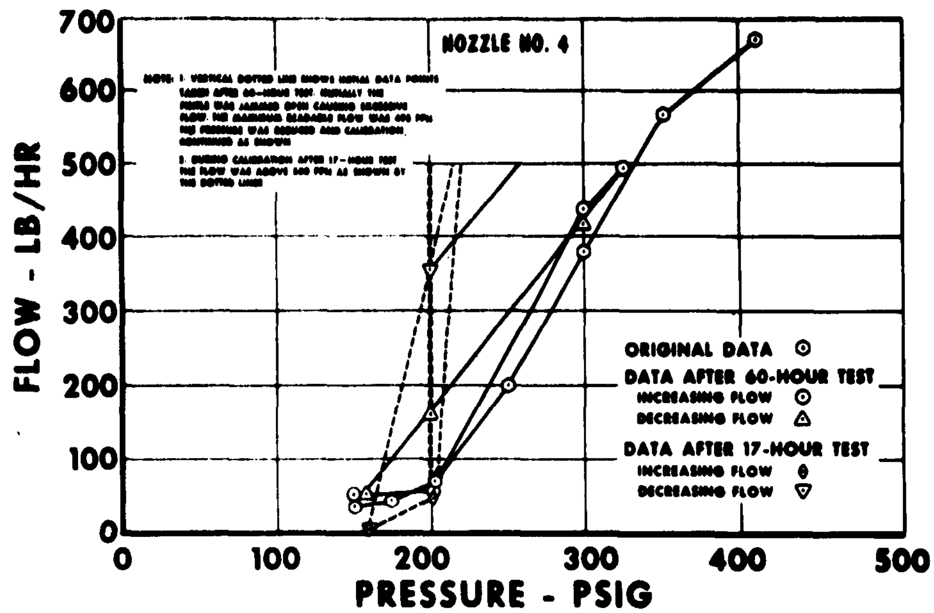


Figure 35 Calibration Data, Variable Area Dual Orifice Nozzle (Nozzle No. 4) ***FD 3031***

This nozzle is shown disassembled in figure 36. It should be noted that fungus and lint have bridged one of the openings at the inlet to the nozzle metering section. The condition of the secondary orifice is shown in figure 37. Note the cotton linters that have accumulated on the pintle. The two openings leading to the primary orifice were plugged – one 100 percent, the other approximately 80 percent.

Salt water caused considerable corrosion on the spring retainer and adjusting shims as shown in figure 37. The retainer was pitted severely and about 75 percent of one shim was corroded away. These pieces are made of martensitic stainless steel, which has only fair corrosion resistance to salt solution. Austenitic stainless steel that has a good corrosion resistance would be a better material for service with contaminated fuel.

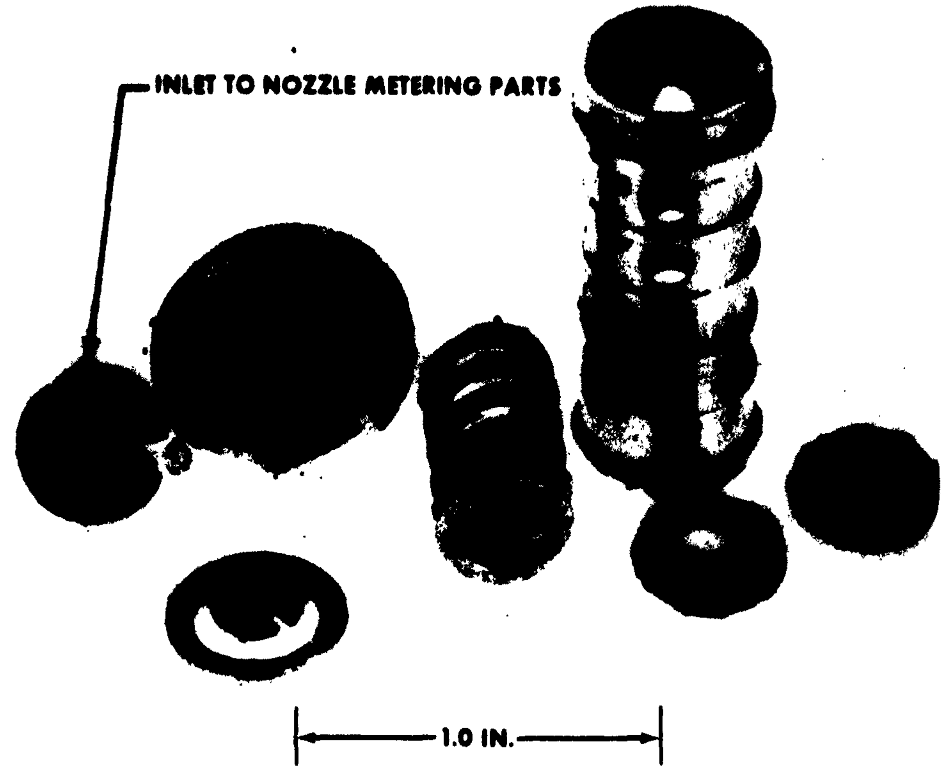


Figure 36 Variable Area Dual Orifice Nozzle (Nozzle No. 4)— Disassembled after Test *FE 18360*

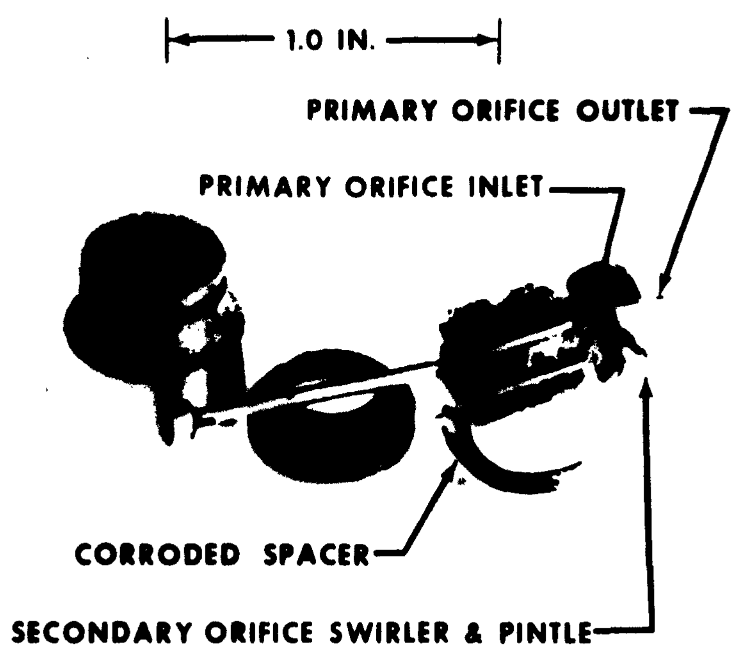


Figure 37 Enlarged View of Secondary Orifice Swirler and Pintle (Nozzle No. 4) *FE 18358*

e. Nozzle No. 5 (Variable Area Nozzle)

A schematic of this nozzle is shown in figure 38. As shown in figure 39, this nozzle was affected by contaminants shortly after the test started. During the first part of the test, contaminant particles were apparently caught between the seat and the pintle because the flow was high and heavy streaks occurred in the spray pattern. These heavy streaks continued during most of the test.

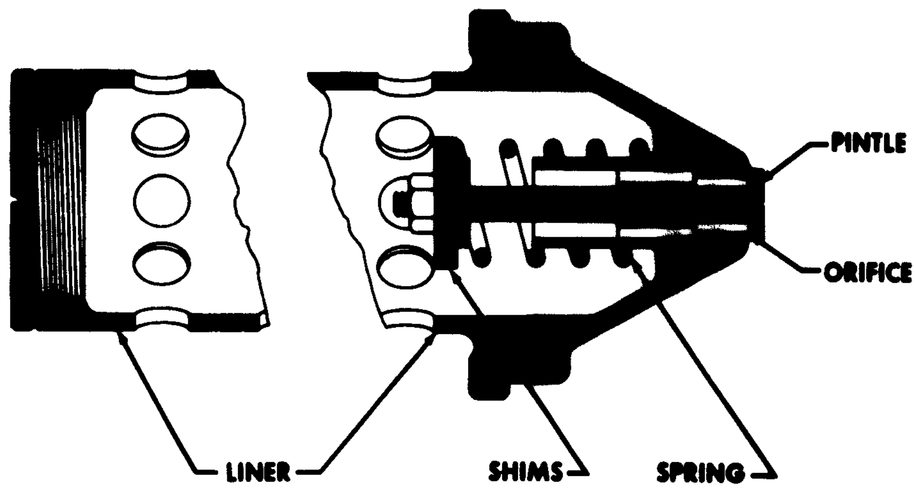


Figure 38 Variable Area Nozzle Schematic (Nozzle No. 5) *FD 3170*

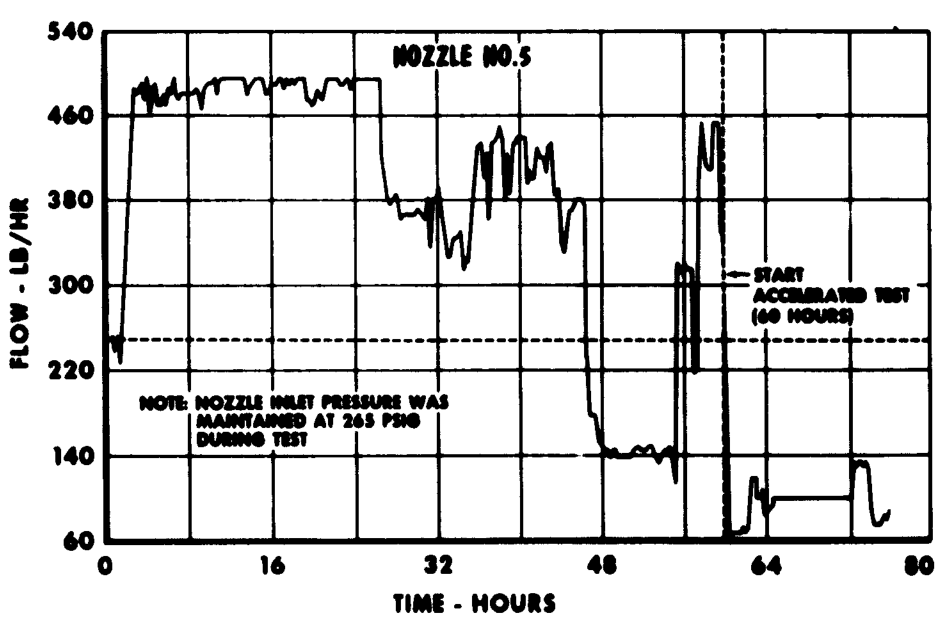


Figure 39 Fuel Flow Variation, Variable Area Nozzle (Nozzle No. 5) *FD 3527*

During the accelerated test the flow from this nozzle dropped much below normal and remained there for the rest of the test. It was not possible to clear the nozzle of contaminants by reducing the pressure.

The calibration data are presented in figure 40. The data taken after the 60-hour test indicated that contamination is holding the pintle open, since a higher than normal flow was measured at pressures up to 300 psig. At 400 psig, the contaminant accumulation at another location in the nozzle limited the flow to a value below normal, as shown. Upon decreasing the pressure, the pintle was held open widely, as demonstrated by the high flow rate. The data taken after the accelerated test illustrate that the nozzle was severely plugged.

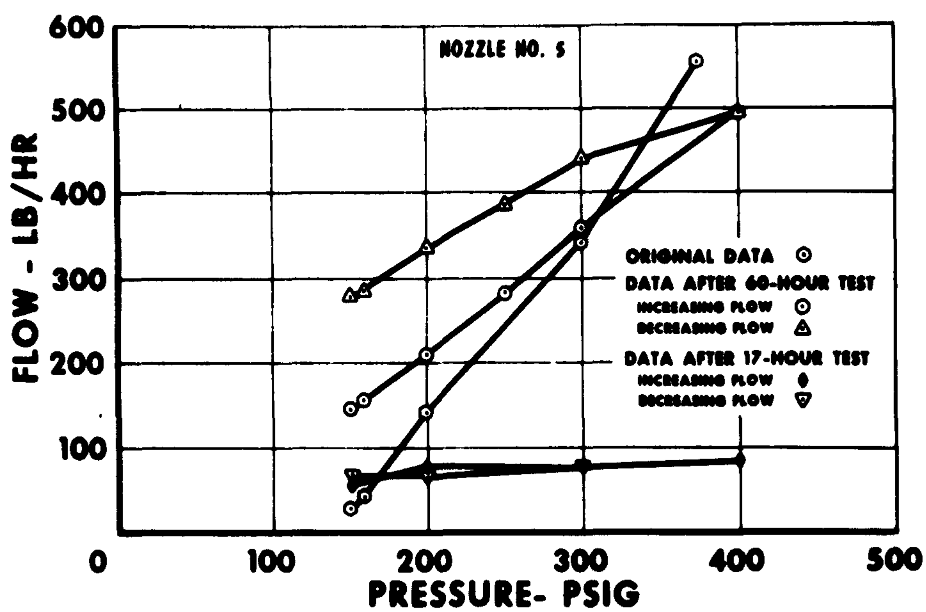


Figure 40 Calibration Data, Variable Area Nozzle (Nozzle No. 5) *FD 3029*

This nozzle is shown disassembled in figure 41. The passages leading to the variable area outlet orifice are completely plugged. (This part of the nozzle is in the nozzle cap and cannot be seen in figure 41.)

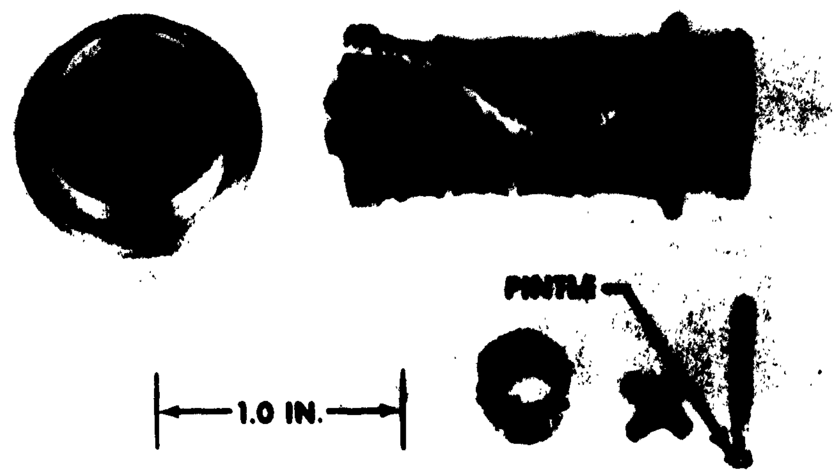


Figure 41 Variable Area Nozzle (Nozzle No. 5)—Disassembled after Test **FD 3168**

f. Nozzle No 6 (Spill Nozzle)

A schematic of this nozzle is shown in figure 42. This nozzle was relatively unaffected by contaminants during the first 60-hour test. During the accelerated test, flow dropped from 740 lb/hr to 180 lb/hr. Examination of the nozzle after disassembly revealed that no contamination existed at any location in the nozzle. However, it was found that erosion had increased the diameter of the swirl chamber approximately ⅛ inch, and in addition had roughened the internal walls of this chamber. Also, a small hole, approximately the same size as the inlet swirl chamber holes, had been eroded through the swirl chamber wall. This hole is shown in figure 43. Because this hole enters the chamber radially and not tangentially (as the original holes do), the turbulence created in the swirl chamber limits the fuel flow to a value below normal for the same inlet nozzle pressure. This fact explains the discrepancy in the calibration data presented in figure 44.

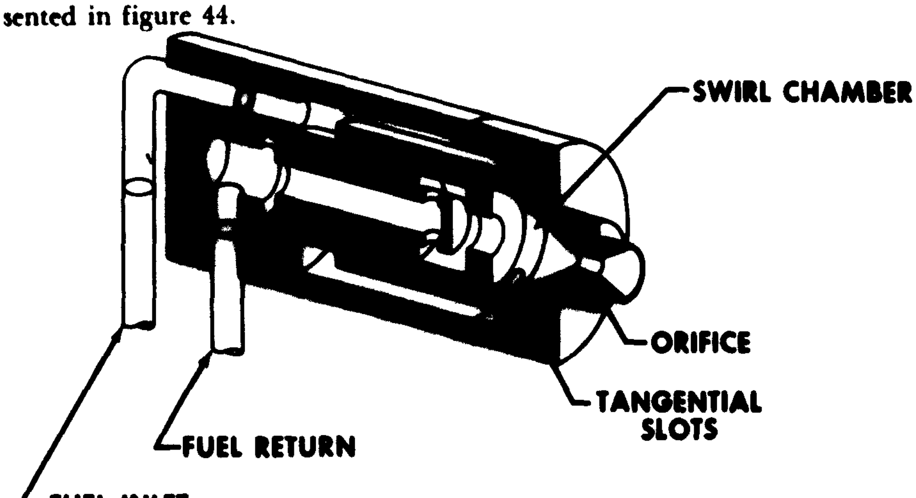


Figure 42 Spill Nozzle Schematic (Nozzle No. 6) **FD 3175**

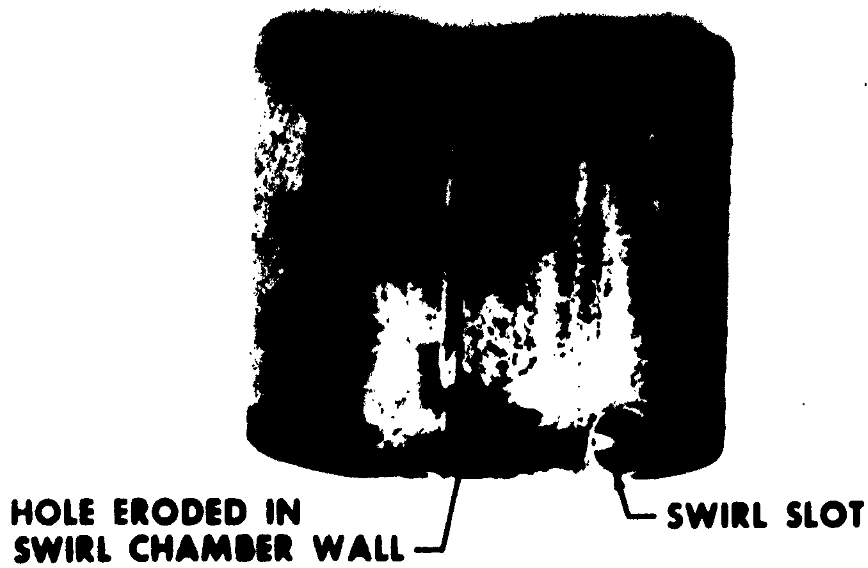


Figure 43 Spill Nozzle (Nozzle No. 6) showing Erosion in Swirl Chamber Wall **FE 19062**

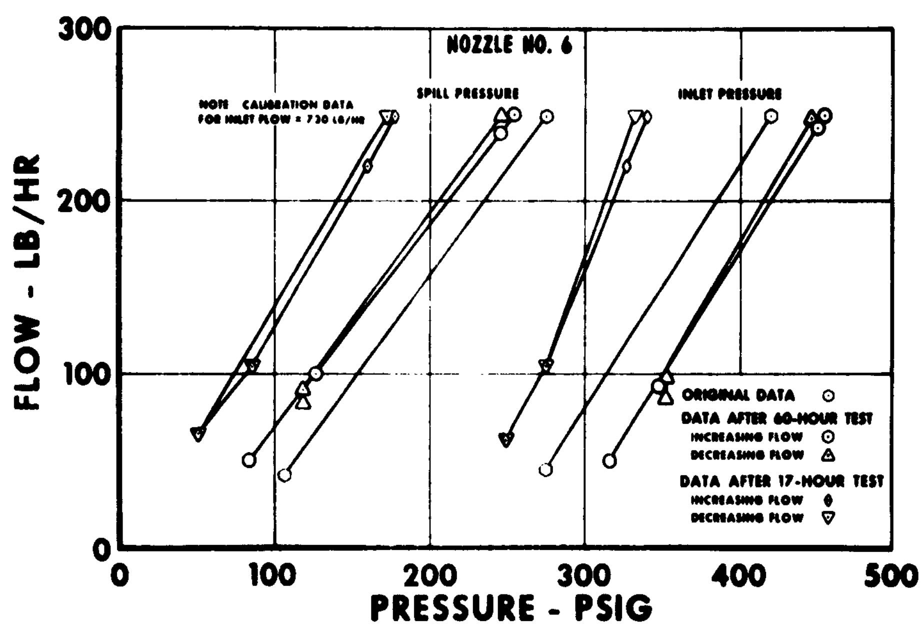


Figure 44 Calibration Data, Spill Nozzle (Nozzle No. 6) **FD 3547**

3. *FINAL NOZZLE SPRAY TEST*

In a final nozzle spray test, four nozzles were evaluated during a 60-hour contaminated-fuel spray test: a dual orifice nozzle that was designed by Delavan Manufacturing Company to spray contaminated fuel, and three variable area dual orifice nozzles. The three modified variable area dual orifice nozzles are the same as the variable area dual orifice nozzle (Nozzle No. 4) that was evaluated during the comparative nozzle spray tests. The nozzles tested are described in table VII.

TABLE VII. NOZZLE DESCRIPTION

Nozzle No.	Description
1	A Delavan-designed dual orifice nozzle for spraying contaminated fuel
2	A variable area dual orifice nozzle with a 320-micron clearance between the pintle and the secondary swirl chamber diameter, and with a 350-micron screen filter
3	A variable area dual orifice nozzle that is identical to Nozzle No. 2, except for the filter
4	A variable area dual orifice nozzle with a 1000-micron clearance between the pintle and the swirl chamber diameter

The four nozzles were tested simultaneously in the same equipment used during the comparative nozzle spray tests.

The following paragraphs describe the nozzles, their performance during the test, and the results of post-test investigations.

a. *Dual Orifice Nozzle (Delavan designed)*

The nozzle design is shown in figure 45. The important dimensions of this nozzle are listed in Table VIII. The thickness of the annulus between the primary and secondary orifices is 0.028 inch; both primary and secondary orifice slots are square.

TABLE VIII. SPECIFICATIONS FOR DELAVAN NOZZLE

	Primary	Secondary
Outlet orifice diameter, inch	0.034	0.125
No. of swirl slots	1	8
Area of a swirl slot, in^2	0.0004	0.0009
Dimension of a square slot with the same area, inch	0.020	0.030

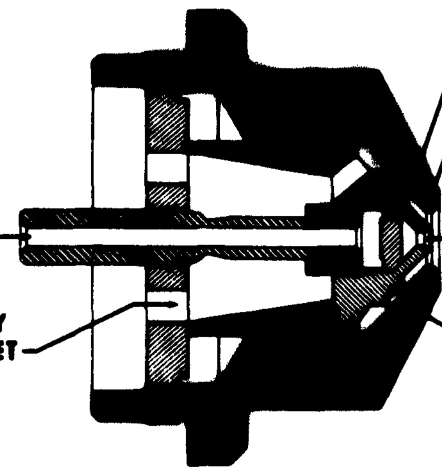


Figure 45 Dual Orifice Nozzle (Delavan Design) *FD 3491*

The flow schedule of this nozzle is the same as that for the variable area dual orifice nozzles. Since a flow divider valve was not supplied with the nozzle, the primary and secondary orifices were tested separately. A pressure of 240 psig was maintained on the primary orifice and 200 psig was maintained on the secondary orifice.

Post-test examinations revealed that the primary orifice ruptured during the test and that a portion of it was blocking the outlet of the secondary orifice. This fact probably caused the poor performance that is described in the following paragraph.

The nozzle flowed poorly soon after the test began. The primary never flowed properly after the initial calibration. After only 12 hours of testing, the flow dropped more than 30 percent. A distorted spray pattern was common throughout the test. The nozzle was removed from the test bench after 47 hours. The calibration curves in figure 46 indicate the loss in flow that this nozzle experienced every 12 hours.

b. Variable Area Dual Orifice Nozzle With A 350-Micron Filter (320-Micron Clearance)

This nozzle was the same type of nozzle that was tested in the comparative nozzle spray test, except that the radial clearance between the pintle and the swirl chamber of the secondary orifice was increased from 125 to 320 microns. This nozzle contained a 350-micron screen at the nozzle inlet.

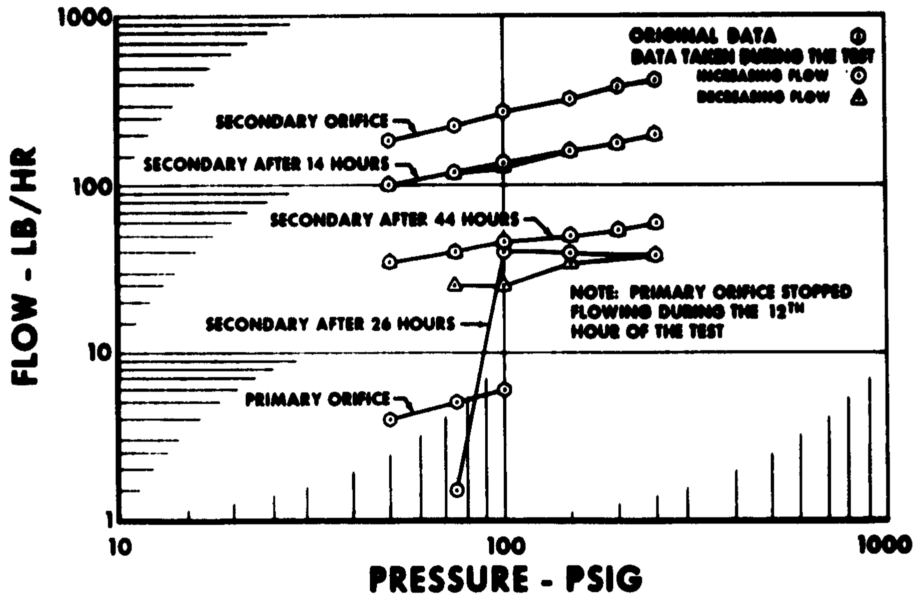


Figure 46 Calibration Data, Dual Orifice Nozzle (Delavan Design) *FD 3387*

The nozzle flowed extremely well during the 60-hour test; there were only small fluctuations in fuel flow as shown in figure 47. It was not necessary to clear the nozzle periodically as it was during the comparative nozzle spray test. As shown in figure 48, there was excellent agreement of the calibration data before and after testing.

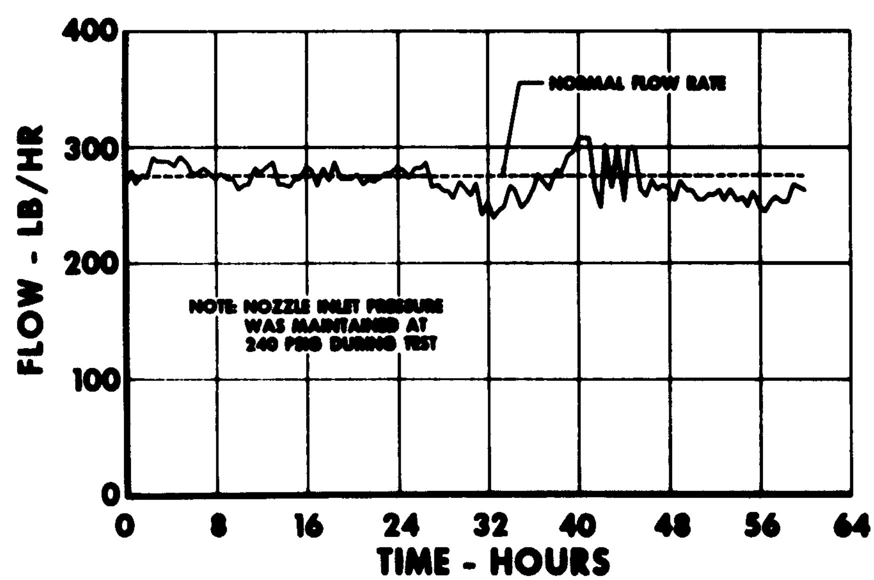


Figure 47 Fuel Flow Variation, Variable Area Dual Orifice Nozzle with 350-Micron Filter (320-Micron clearance) *FD 3505*

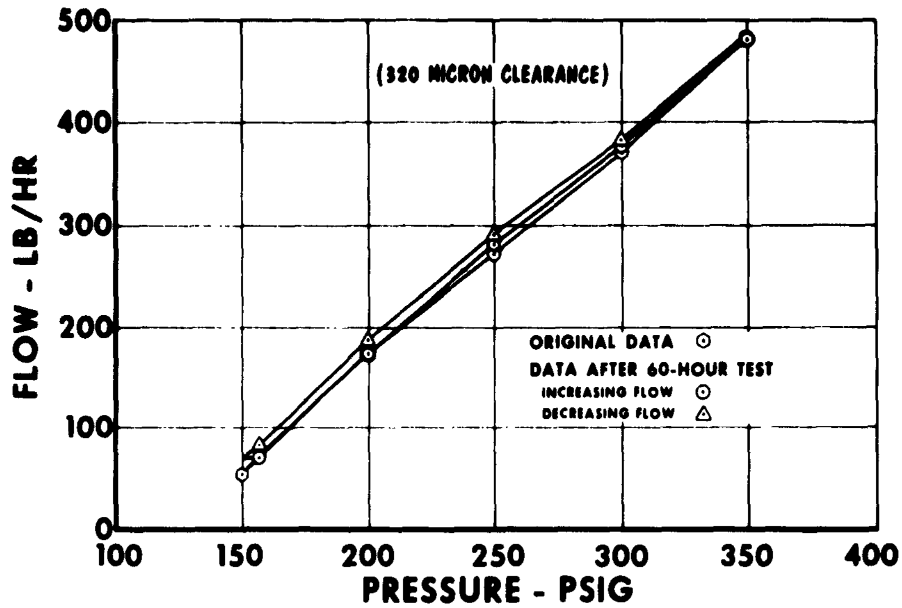


Figure 48 Calibration Data, Variable Area Dual Orifice Nozzle with 350-Micron Filter (320-Micron Clearance) *FD 3388*

The good performance of this nozzle illustrates another important point — the minimum required clearance between moving parts. In this nozzle the clearance between parts A and C of figure 33 is 0.00018 inch (4.5 microns). The particles of contaminant less than 5 microns either did not get between the parts or their presence did not cause seizure of the poppet. With this minimum dimension established, it appears that the clearance between moving parts should be less than 0.00018 inch or greater than 0.0128 inch.

The nozzle was disassembled after testing. The primary orifice was free of any contaminant deposits. The secondary orifice was relatively free of contaminants as shown in figure 49. It is apparent that the 350-micron screen filtered the large particles from the nozzle, which permitted satisfactory operation for the duration of the test.

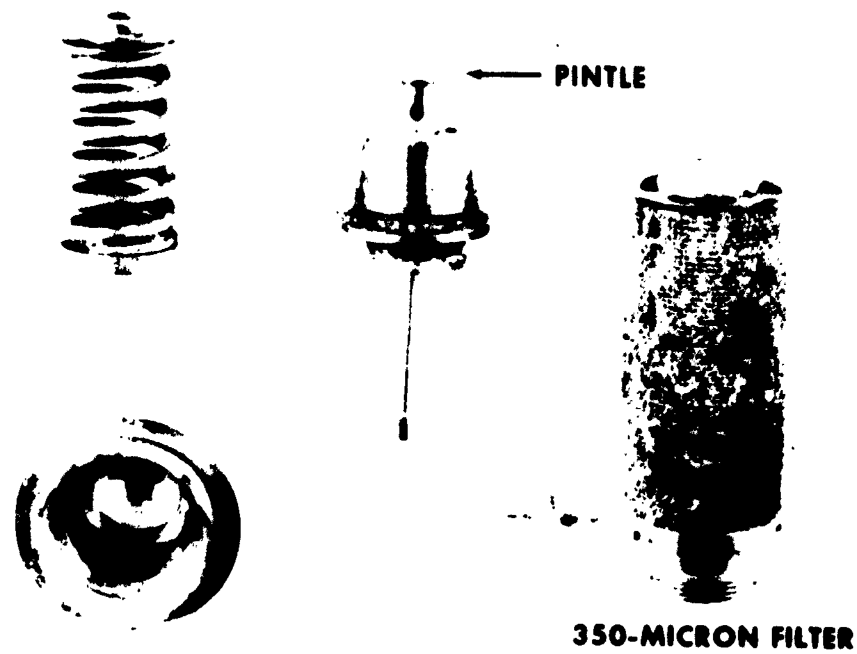


Figure 49 Variable Area Dual Orifice Nozzle with 350-Micron Filter (320-Micron Clearance) *FD 3396*

c. Variable Area Dual Orifice Nozzle (320-Micron Clearance)

This nozzle is identical to the nozzle discussed in b above; it was tested without a screen. The performance of this nozzle was good, and it was not necessary to clear the nozzle by reductions in the inlet pressure. However, some contaminant did accumulate in the nozzle, as it lost approximately 20 percent of its flow during the 60-hour test as shown in figure 50. This is also shown in figure 51, which compares the calibration data before and after testing.

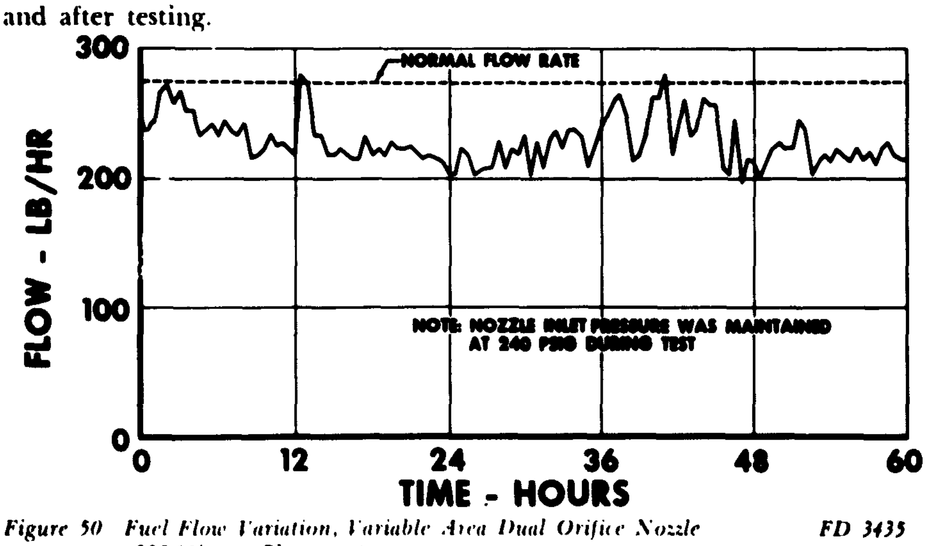


Figure 50 Fuel Flow Variation, Variable Area Dual Orifice Nozzle (320-Micron Clearance) *FD 3435*

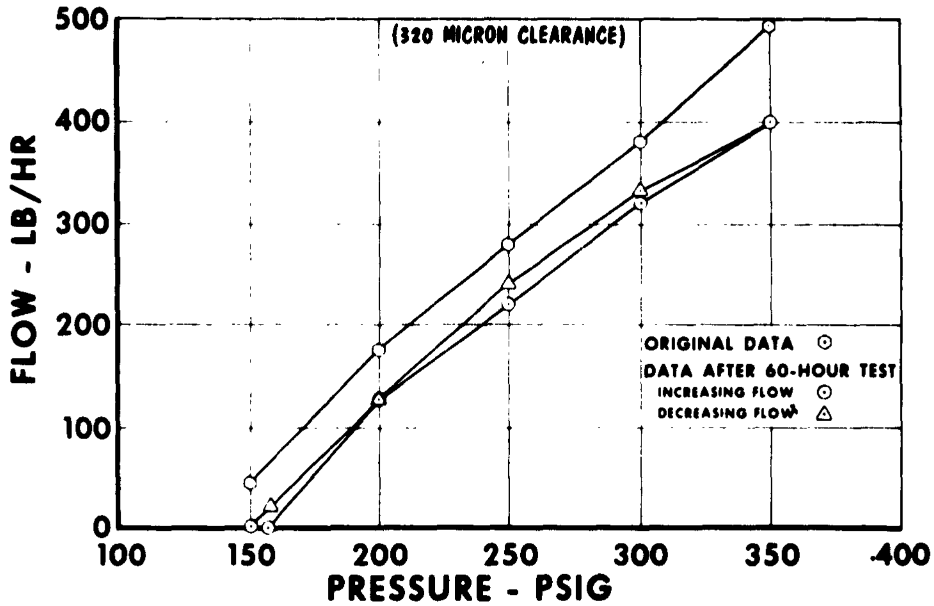


Figure 51 Calibration Data, Variable Area Dual Orifice Nozzle (320-Micron Clearance) *FD 3385*

Examination of the disassembled nozzle revealed that the primary orifice had clogged. The secondary orifice contained some linters and fungus, although accumulation was not severe. A photograph of the nozzle parts is shown in figure 52.

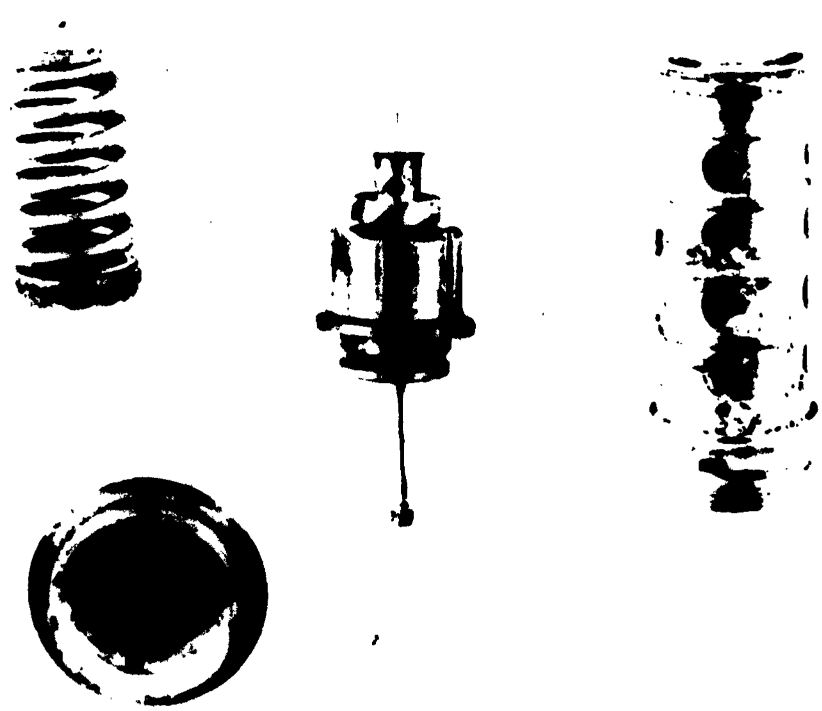


Figure 52 Variable Area Dual Orifice Nozzle (320-Micron Clearance) ***FE 19381***

d. Variable Area Dual Orifice Nozzle (1000-Micron Clearance)

This nozzle is the same nozzle type as those discussed in a and b above, except that the clearance between the primary and secondary body was increased to 1000 microns.

The flow of the nozzle dropped approximately 25 percent during the 60-hour test as indicated in figure 53 and on the calibration curves in figure 54. It was found during post-test examinations that the primary orifice was plugged; this is the cause for the drop in flow. A photograph of the nozzle parts is shown in figure 55.

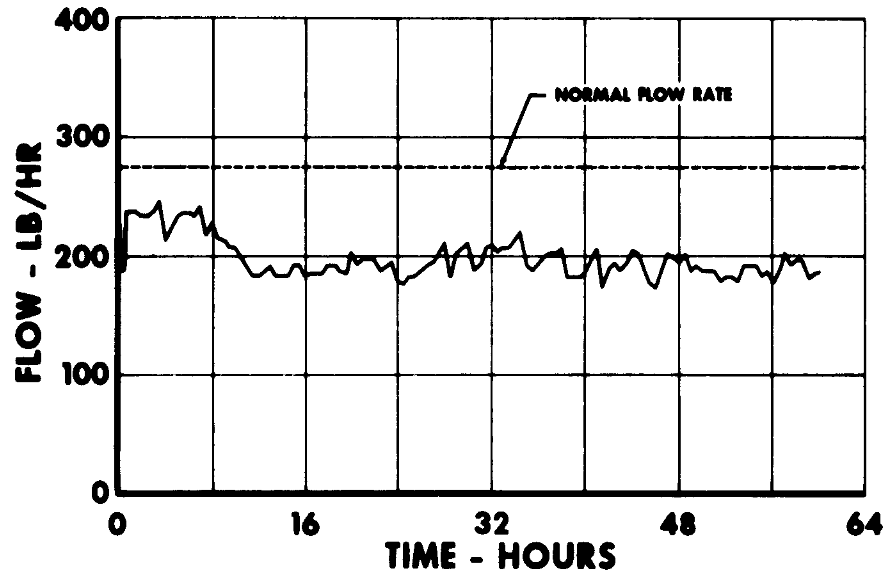


Figure 53 Fuel Flow Variation, Variable Area Dual Orifice Nozzle (1000-Micron Clearance) *FD 3523*

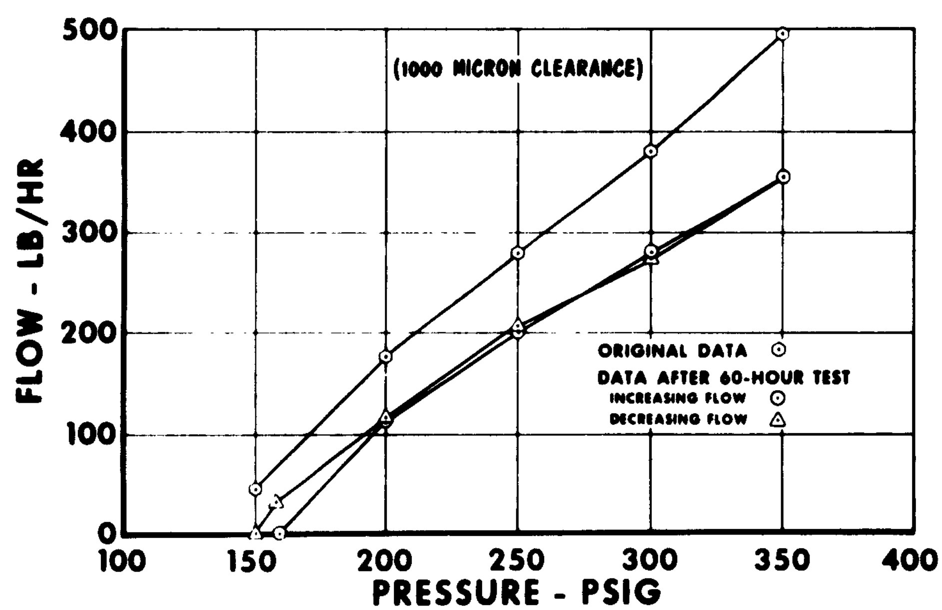


Figure 54 Calibration Data, Variable Area Dual Orifice Nozzle (1000-Micron Clearance) *FD 3386*

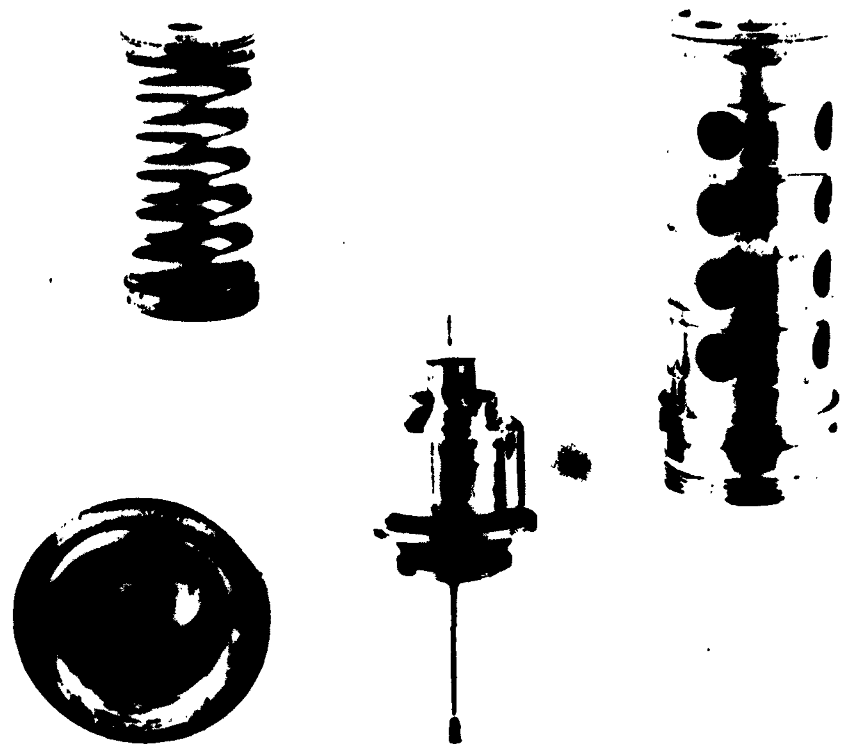


Figure 55 Variable Area Dual Orifice Nozzle (1000-Micron Clearance) **FE 19380**

e. Results

Comparison between the results obtained with the modified variable area dual orifice nozzles and those obtained with nozzle No. 4 of the comparative nozzle spray test indicates that improvements can be made to nozzles to improve their performance when spraying contaminated fuel. It also can be concluded that a screen at the nozzle inlet is very important. It is believed that a central filter is not as effective as a filter at the nozzle inlet because contaminants apparently agglomerate after going through the central filter. This results in large particles of contaminant that are capable of clogging nozzles. A filter at the nozzle inlet prevents these larger particles from entering the nozzle and the contaminants do not have sufficient time to agglomerate while passing through the nozzle.

4. FUEL NOZZLE CLUSTER CONDITIONING TESTS

Two nozzle cluster conditioning tests were made with contaminated fuel. Both of the clusters contained variable area dual orifice nozzles (the same design as nozzle No. 4 of the comparative nozzle spray tests). The first of these was made to check the performance of the test stand; it was made with fuel contaminated with MIL-F-5007B only (no fungus was added).

The second of these tests was a 60-hour test to condition a fuel cluster for use during the burner can test program. The pressure and flow variations for these two tests are presented in figures 56 and 57.

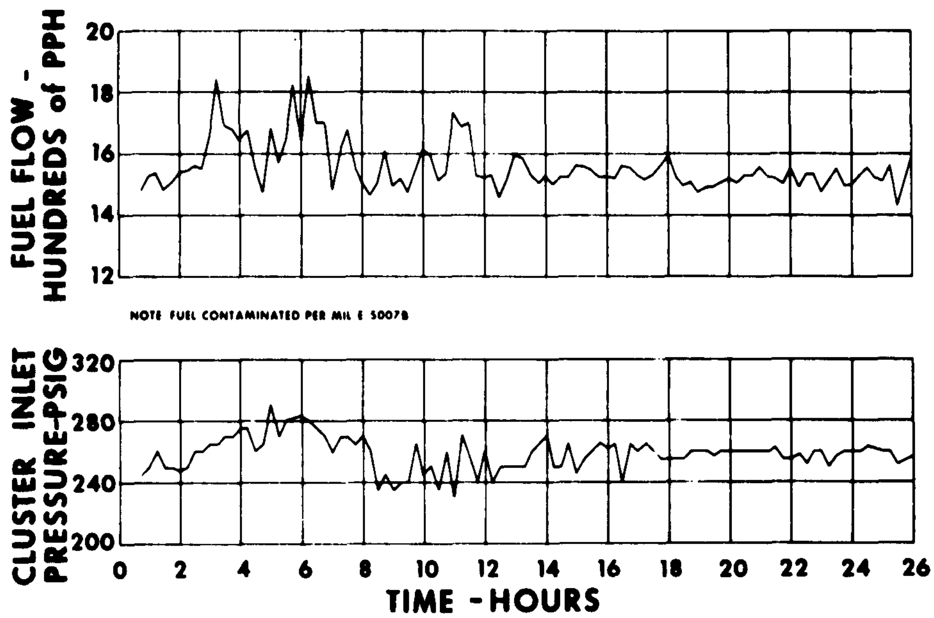


Figure 56 Fuel Nozzle Cluster Performance *FD 2572*

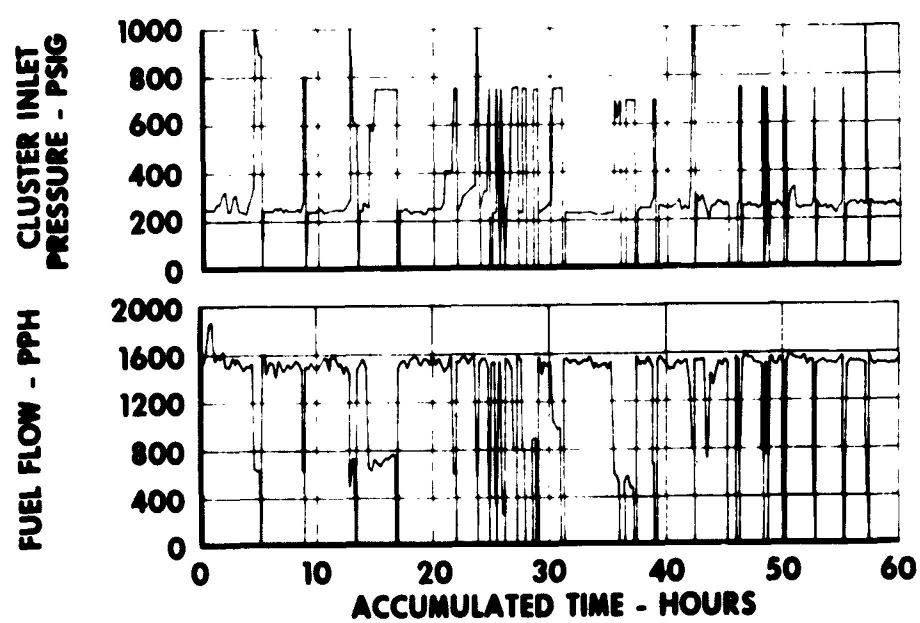


Figure 57 Fuel Nozzle Cluster Performance *FD 2668*

As shown in figure 56, there were only small variations in flow and pressure during the first test, most of which were caused by a defective metering valve. The nozzles performed well, with only slight streaks detected in the spray pattern. The nozzles were also self-cleaning; as the contaminant accumulated in the nozzles, the pressure would increase slightly. Then the contaminant was flushed from the nozzle, and the pressure returned to normal.

With the fungus as a contaminant in the second test, the flow decreased and the pressure increased abruptly and frequently. Also, the spray patterns were severely affected. In some instances, the spray angle decreased from 80 degrees to as low as 40 degrees. Also there were heavy streaks in the spray pattern. On several occasions, the entire flow from one nozzle was flowing in three or four large streaks.

From the results of these two tests it is concluded that the fungus solution can affect fuel nozzle performance severely.

SECTION VI

BURNER CAN TESTS

A. GENERAL

Burner can tests were made with contaminated fuel as fulfillment of Work Statement Item 5 of the contract. The purpose of these tests was to determine the effects of the contaminated fuel on burner can performance. Both variable area nozzles and variable area dual orifice nozzles (the same design and size as those evaluated during the nozzle spray tests) were used in clusters of six. The variable area dual orifice nozzles were of the modified type (the clearance between the pintle and the secondary swirl chamber was [illegible] microns).

The effect of contaminated fuel on burner can performance was obtained by comparing the contaminated fuel performance data with burner can base data obtained with clean fuel. With each cluster of fuel nozzles, a burner can calibration test with clean fuel was made; then the cluster was tested with contaminated fuel.

This section of the report includes (1) conclusions, (2) a description of the test stand, (3) a description of the burner can tests and (4) the burner can test results.

B. CONCLUSIONS

1. Combustion efficiency is not affected seriously by contaminated fuel. The maximum decrease that could be attributed to contaminated fuel was 8 percent. Although this decrease is significant, because the combustion efficiency did not remain at a low value for an extended period of time, the occurrence of this decrease is not considered to be serious.

2. The most serious effect caused by contaminated fuel is a distorted burner can exit temperature distribution. The temperature variation at the burner can exit increased from 420°F during run 4 (made with clean fuel) to 1065°F during run 5 (a contaminated fuel run).

3. The contaminated fuel caused the critical burner can temperature rise to decrease. During run 5 the critical temperature rise dropped from 1400° to 700°F.

4. Operation with contaminated fuel did not cause hot spots in the burner can wall at any time during the test.

5. Contaminated fuel did not significantly affect the fuel-to-air ratio operating range of the combustor.

6. The results of the tests support the findings of the comparative nozzle spray test. The variable area nozzle is not suitable for use with contaminated fuel. The modified variable area dual orifice nozzle works satisfactorily with contaminated fuel.

C. DISCUSSION

1. DESCRIPTION OF THE TEST STAND

The burner can tests were made in a test stand that was built to duplicate the airflow pattern and conditions that the burner can would experience in an engine. Consequently the results obtained during the burner can tests are those that could be expected to occur in an engine burning contaminated fuel.

The contaminant addition system used in the nozzle spray tests was utilized in the burner can program. This system was described in Section V of this report. The system was connected in parallel with the test stand fuel pump. This arrangement permitted the utilization of the stand pump for the clean fuel tests, and thus ensured the validity of the base data.

2. BURNER CAN TEST PROCEDURES

Eighteen burner can tests were made during the program; these are described in table IX.

TABLE IX. DESCRIPTION OF BURNER CAN TESTS

Run No.	Nozzle Type	Test Type	Fuel
1	Variable Area Dual Orifice (VADO)	Standard Burner Calibration	Clean
2	VADO	Cycle	Contaminated
3	VADO	Standard Burner Calibration	Contaminated
4	Variable Area	Standard Burner Calibration	Clean
5	Variable Area	Cycle	Contaminated
6	VADO	Standard Burner Calibration	Clean
7	VADO	Standard Burner Calibration	Contaminated
8	VADO	Standard Burner Calibration	Clean
9	VADO	Heat Soak	Contaminated
10	VADO	Standard Burner Calibration	Contaminated
11	VADO	Standard Burner Calibration	Clean

TABLE IX. DESCRIPTION OF BURNER CAN TESTS (Contd)

Run No.	Nozzle Type	Test Type	Fuel
12	VADO	Endurance Run (10 Hours)	Contaminated
13	VADO	Cycle	Contaminated
14	VADO	Standard Burner Calibration	Clean
15	VADO	Standard Burner Calibration	Clean
16	VADO	Standard Burner Calibration	Clean
17	VADO	Cycle	Contaminated
18	VADO	Standard Burner Calibration	Contaminated

All of the runs made with clean fuel except runs 14 and 15 are base data runs. Each is used with the set of contaminated fuel runs that immediately precede the clean fuel run. For example, run 1 is the base data run for runs 2 and 3, and run 4 is the base data run for run 5. Runs 14 and 15 were made with the fuel cluster that was conditioned by a 60-hour cold-flow test with contaminated fuel which was described in Section V of this report. During all of the runs the following variables were recorded:

a. Airflow
b. Burner can inlet total pressure
c. Burner can inlet static pressure
d. Burner can inlet air temperature
e. Fuel flow
f. Cluster inlet fuel pressure
g. Cluster inlet fuel temperature
h. Exhaust stack air temperature
i. Traverse air temperature readings at the burner can exit
j. Traverse total pressure readings at the burner can exit
k. Radiation intensity at the burner can exit
l. Burner can wall temperatures.

Measurement of the foregoing parameters provides information for determining the following burner can performance parameters.

a. Combustion Efficiency—Obtained by the ratio of actual-to-ideal temperature rise. The stack temperature (the temperature obtained downstream of the burner away from the effects of visible radiation) is used rather than the average temperature at the burner can exit to compute the actual temperature rise.

b. Temperature Distribution at the Burner Can Exit—Obtained by use of a moveable rake. This rake consists of seven thermo-

couples equally spaced radially in the annulus between the inner and outer radii of the burner can exit. By rotating the rake, a complete map of the air temperature at the burner can exit can be obtained.

c. Critical Temperature Rise–The ability of a burner to complete combustion within itself is an important performance feature. Flame length is measured by observing the intensity of radiation at the burner exit with a radiation analyzer. The critical temperature (the temperature at which the flame reaches the exit of the burner can) can be determined by extrapolating a radiation intensity versus temperature rise curve.

d. Burner Can Wall Temperatures–Monitored to determine whether there are any hot spots on the burner can wall.

e. Lean Blowout–The fuel-to-air ratio at which flame extinguishment occurs can be obtained by reducing the fuel flow while maintaining the air conditions constant. (Rich blowout limits were not determined because of test stand temperature limitations.)

f. Pressure Loss–Burner can pressure loss was calculated. However, since this parameter is dependent primarily on burner can geometry, it should not be affected by contaminated fuel unless a hole is burned in the burner can wall.

3. *DESCRIPTION OF THE BURNER CAN TESTS*

The types of burner can tests made during the program are (1) standard burner can tests, (2) cycle tests, (3) a heat soak test and (4) an endurance test. These are described in the following paragraphs.

a. Calibration Tests

With the airflow and burner can inlet temperature and pressure set, the stack temperature was increased (by increasing fuel flow) in increments of 200°F from approximately 1200° to 1600°F. At each step, readings were taken of all the variables defined above, except the traverse rake temperatures and pressures. At 1600°F all instrumentation was read, including the traverse rake readings.

NOTE

In the succeeding pages, the following definitions will be used:

long reading – readings of all the variables listed in C2, preceding

short reading – readings of all the variables listed in C2, preceding, except the traverse rake temperature and pressures.

The stack temperature was then increased to 1800°F, dropped to 1700°F, and subsequently dropped in 200°F steps to 1300°F. At each of these stack temperatures a short reading was taken. The fuel flow was then reduced gradually until blowout occurred. The important achievements of this procedure are: (1) the variations of combustion efficiency with burner can temperature rise, (2) an estimate of critical temperature rise, and (3) temperature distribution at the burner can exit at a high stack temperature.

b. Cycle Tests

The hysteresis in the fuel flow schedule that occurs as a result of contaminant accumulation in the nozzle will probably cause hot spots at the burner can exit. Therefore, cycle tests were made in which the fuel flow was raised and lowered frequently to see if any hysteresis would occur. In this test, as in the burner calibration tests, the inlet air conditions and the airflow were maintained constant during the entire run.

In the cycle tests following lightoff, short readings were taken at 1400°, 1600°, and 1800°F. Then the stack temperature was reduced to 1600°F where a long reading was taken: the fuel flow was then dropped by approximately 10 percent. If the inlet cluster pressure was abnormally low or high and remained there for over one minute, a long reading was taken. Following this, the stack temperature was dropped to 1400°F and a short reading was taken. Then the stack temperature was increased to 1600°F where another short reading was taken. The entire procedure was then repeated. In run 2 this procedure was repeated 10 times. Periodically during the test, and at the conclusion of the test, long readings were taken at a 1600°F stack temperature. At the conclusion of the test, the flow was gradually decreased until blowout occurred.

c. Heat Soak Test

A heat soak test was made to determine the effect of both fuel gum deposits and contamination deposits in the nozzle on burner can performance. During this test the burner can air inlet temperature was maintained at 900°F. This temperature is sufficiently high to vaporize the fuel and cause gum deposits in the nozzle. During the test, the stack temperature was increased to 1800°F, where a short reading was taken. This stack temperature was maintained for five minutes, then the fuel flow was shut off and the nozzles allowed to "heat soak" for 10 minutes. At the end of this time, fuel flow was started and the stack temperature was increased to 1800°F, and the cycle repeated. A long reading was taken at the conclusion of the test to determine the resulting effect on temperature distribution at the burner can exit.

TABLE X. RESULTS OF BURNER CAN TEST

Run No.	Contaminant Fuel Condition	Combustion Efficiency Variation	Minimum Allowable Temperature Rise, °F	Burner Can Exit Temperature Variation, °F	Burner Can Skin Temperatures, °F	Fuel Flow at Lean Blowout, lb/hr	Pressure Loss, In. Hg
1	Clean	81 - 88	1200	1306 - 2000	284 - 998	48	5.0
2	Contaminated	89 - 99	820	1329 - 2144	275 - 1115	*	5.2
3	Contaminated	88 - 100	970	1353 - 1958	270 - 990	48	5.2
4	Clean	86 - 93	1368	1407 - 1827	280 - 1035	*	5.4
5	Contaminated	90 - 99	700	1231 - 2296	360 - 1080	*	*
6	Clean	92 - 97	1100	1407 - 1812	260 - 1025	*	5.6
7	Contaminated	91 - 100	1360	1415 - 1855	240 - 1020	26	5.2
8	Clean	81 - 87	1050	1173 - 1757	286 - 1190	*	6.3
9	Contaminated	76 - 83	970	1309 - 1969	280 - 1300	50	6.2
10	Contaminated	85 - 101	*	1444 - 1824	590 - 1310	*	5.2
11	Clean	92 - 95	1190	1373 - 1948	460 - 1190	*	*
12	Contaminated	89 - 99	*	1356 - 2011	565 - 1010	*	5.2
13	Contaminated	90 - 97	*	1375 - 1965	*	*	5.3
14	Clean	91 - 97	1100	1372 - 1827	420 - 1050	30	5.4
15	Clean	85 - 95	1580	1424 - 1779	545 - 1250	80 100	6.1
16	Clean	81 - 88	1380	1390 - 1885	*	*	4.9
17	Contaminated	79 - 89	1200	1335 - 1980	290 - 1045	10 50	5.1
18	Contaminated	83 - 88	1490	1336 - 1926	460 - 1010	60	5.5

* Not Available

d. Endurance Test

An endurance test was made at a stack temperature of 1600°F, this temperature was maintained for 10 hours. During this time, consecutive long readings were taken to observe the effects of contaminants on temperature distribution and combustion efficiency. This test was made because it was believed that during the other tests the fuel flow might be changing enough to keep the nozzles clean. During this test the fuel flow was approximately constant.

4. *RESULTS OF BURNER CAN TESTS*

The results of the burner can tests are summarized in table X. The data obtained for each burner can performance parameter is discussed in detail below.

a. Combustion Efficiency

The combustion efficiency data obtained during the program are presented in figures 58 to 64. In these figures the data obtained with contaminated fuel are compared with those obtained during the base data runs. As shown in figure 58 and 59, the data obtained during the first three contaminated fuel runs (runs 2, 3 and 5) were higher than the base data. This is attributed to a low indicated fuel flow that resulted because the turbine meter that was being used to measure fuel flow to the fuel nozzle cluster was affected by the contaminants. In the later runs a flowrater was used; this meter gave satisfactory results.

The fuel cluster used in runs 14 and 15 was conditioned in cold-flow contaminated fuel tests that were conducted approximately two months before the burner can tests were made. This cluster was calibrated with clean fuel. All of the combustion efficiency data are above 87 percent, as shown in figure 63. This indicates that the combustion efficiency was not reduced even though the cluster was used to spray contaminated fuel for 60 hours and then stored for approximately 2 months before the calibration tests were made.

The combustion efficiency data is summarized in figure 65 where the maximum decrease in combustion efficiency from the base data is presented for each of the contaminated fuel runs. The values were obtained for each of these runs by computing the change in combustion efficiency from the base data for each value of burner can temperature rise. The values presented are the maximum change for each run. The temperatures at which these occurred are shown in parentheses below the data point. The values of combustion efficiency that are above zero indicate that the contaminated fuel data was above the base data.

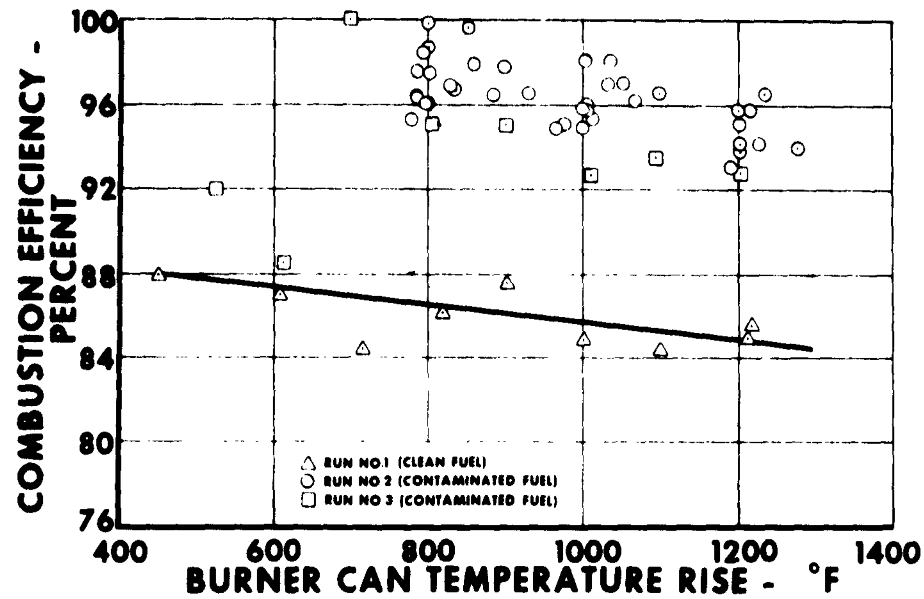


Figure 58 Combustion Efficiency Data Comparison—Runs 2 and 3 with 1 *FD 3568*

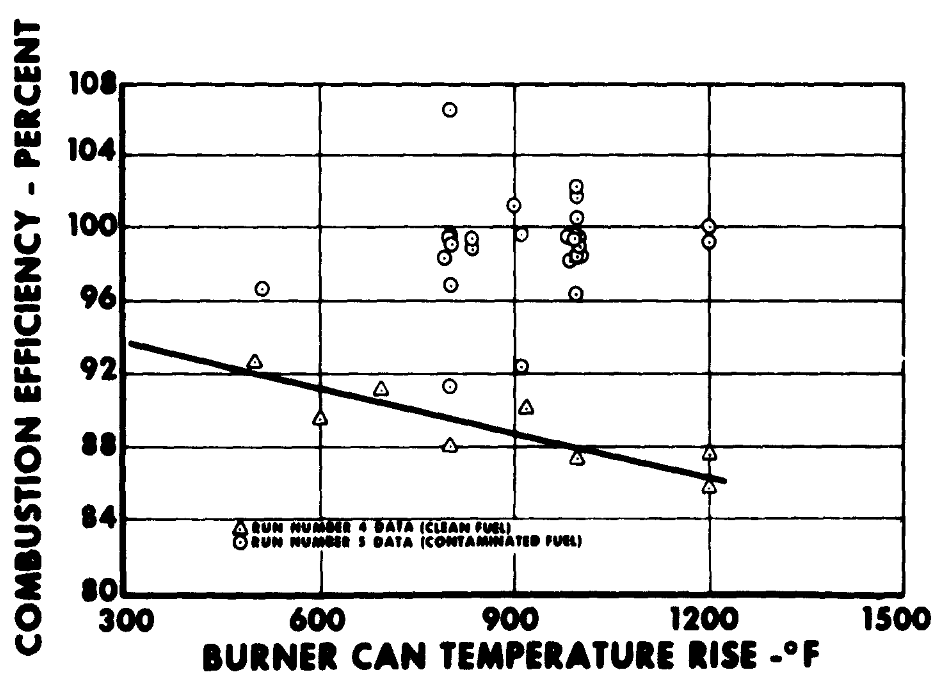


Figure 59 Combustion Efficiency Data Comparison—Run 5 with 4 *FD 3514*

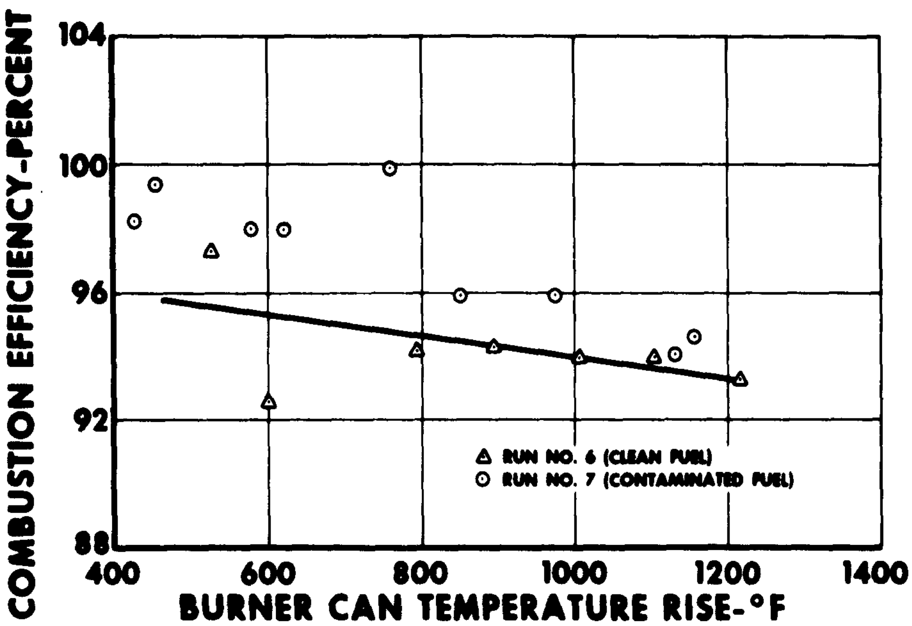


Figure 60 Combustion Efficiency Data Comparison—Run 7 with 6 *FD 3513*

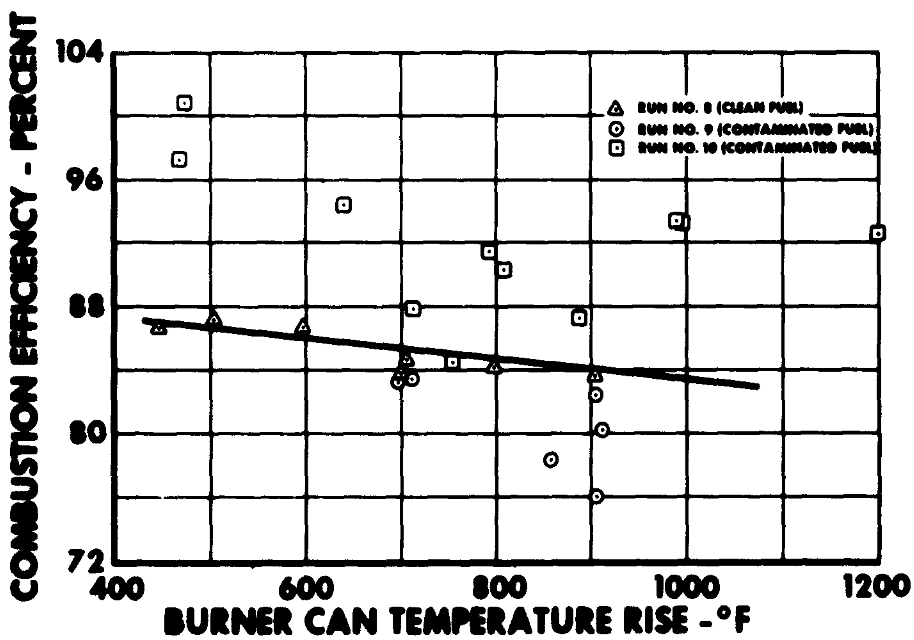


Figure 61 Combustion Efficiency Data Comparison—
Runs 9 and 10 with 8 *FD 3539*

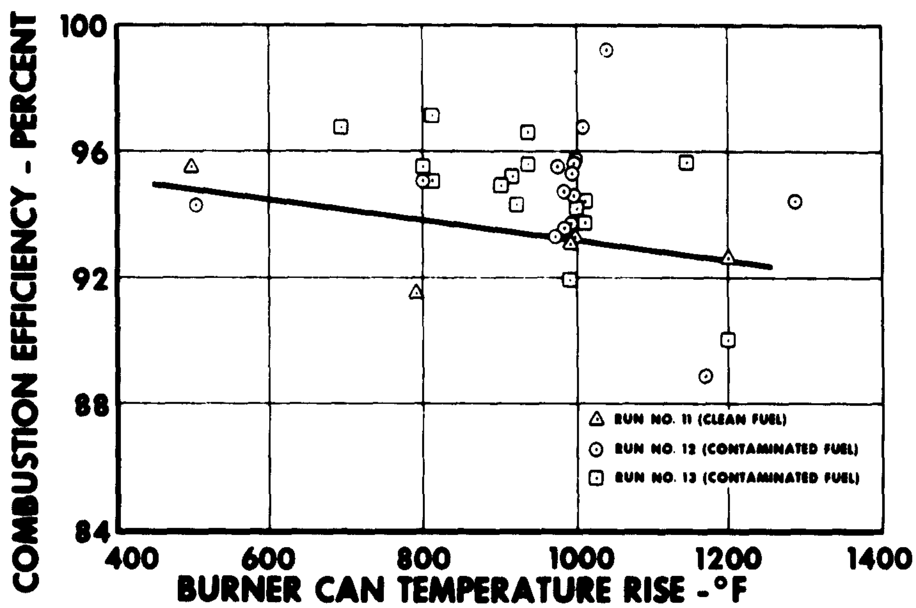


Figure 62 Combustion Efficiency Data Comparison— Runs 12 and 13 with 11 *FD 3552*

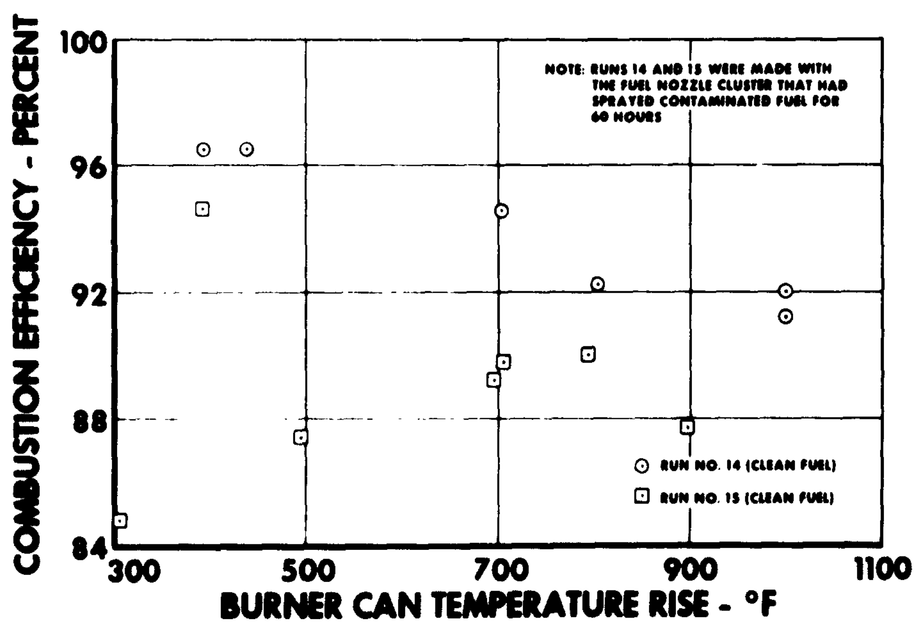


Figure 63 Combustion Efficiency Data—Runs 14 and 15 *FD 3543*

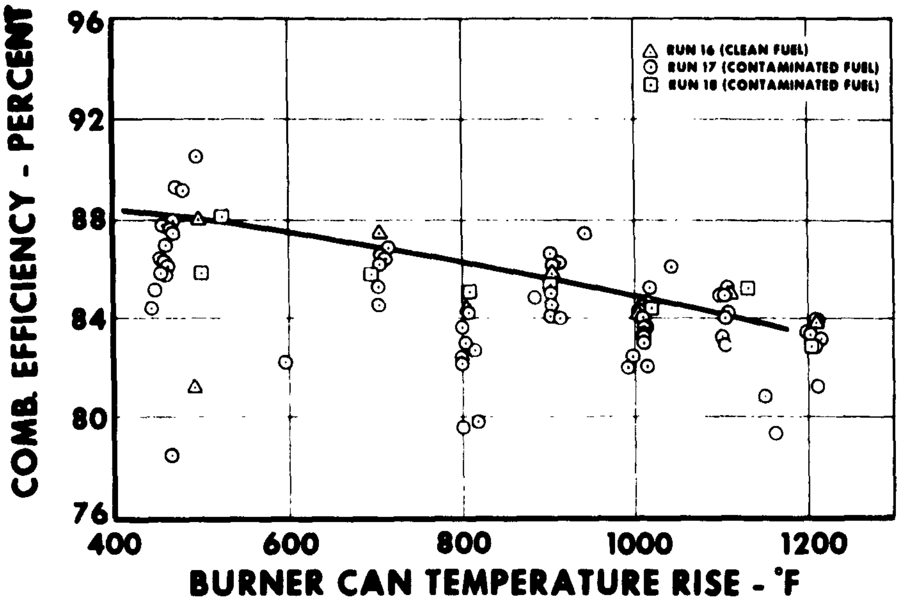


Figure 64 Combustion Efficiency Data Comparison— Runs 17 and 18 with 16 *FD 3562*

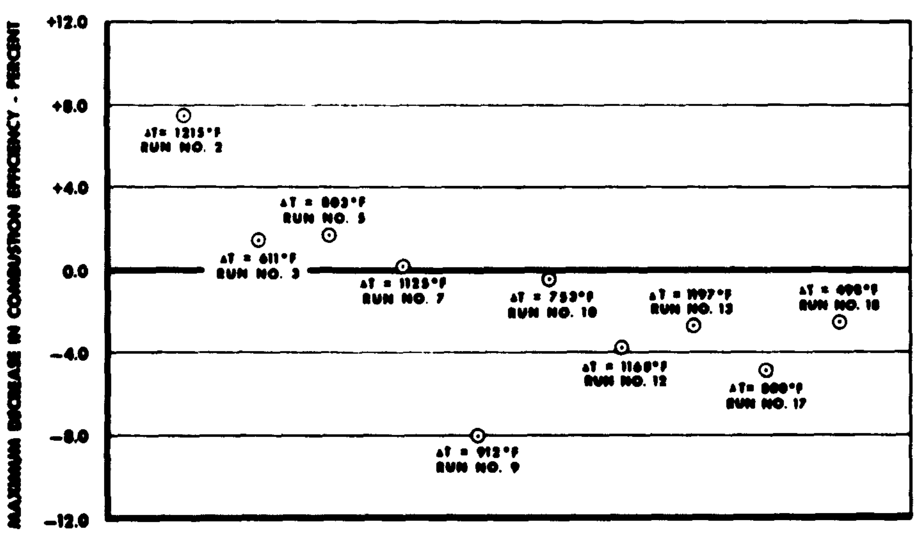


Figure 65 Summary of Combustion Efficiency Data Comparison *FD 3558*

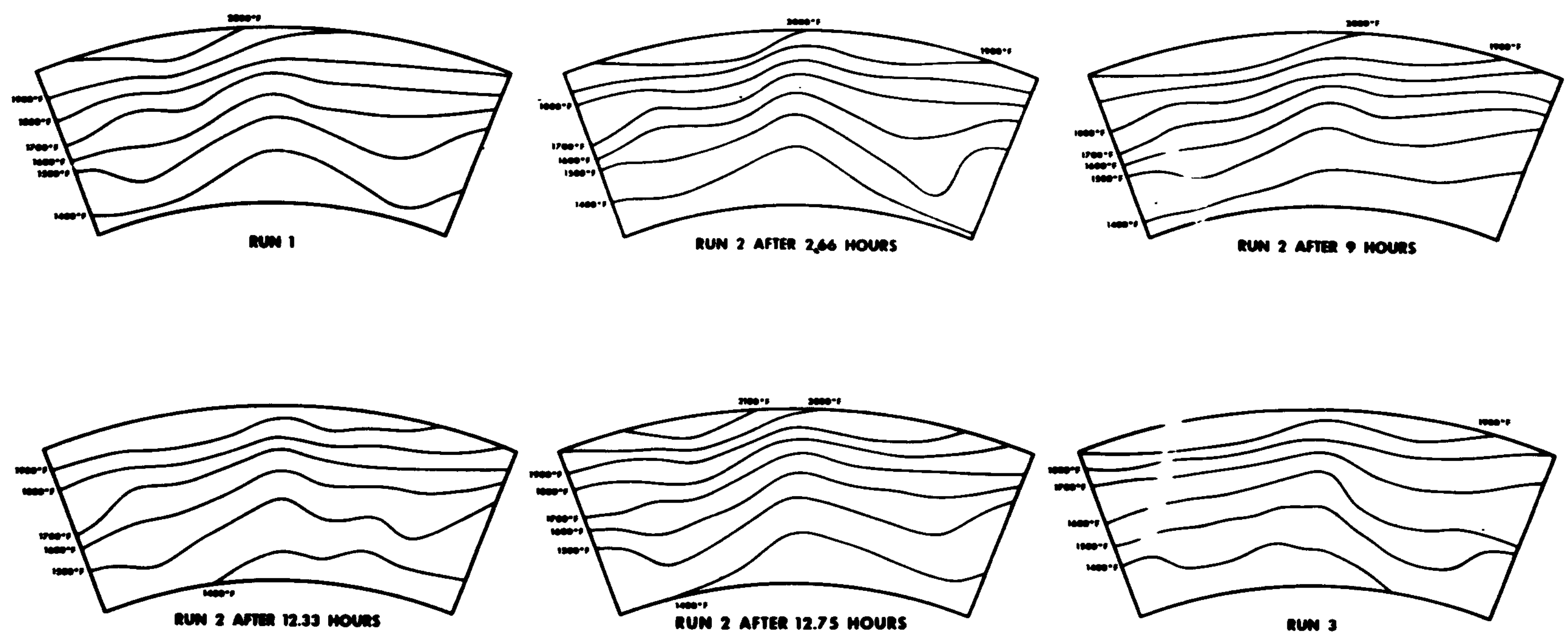
The maximum decrease in combustion efficiency that can be attributed to contaminated fuel is 8 percent. This value was obtained during run 9, the heat soak run. This low efficiency occurred approximately one hour after the start of test. During the remaining eight hours of the test the combustion efficiency was never this low again even though nine relights were made. Because the combustion efficiency did not remain at a low value, this decrease in combustion efficiency, although significant, is not considered to be serious.

b. Temperature Distribution at the Burner Can Exit

The temperature distributions at the burner can exit are presented for each of the runs in figures 66 to 72, inclusive. The temperature maps that were obtained with the traverse rake are presented in sequence as they were observed during the tests. Therefore, the effects of contamination accumulation are shown as the tests progressed.

The most distorted temperature maps were obtained during run 5. This run was made with variable area nozzles, which are the same type and size as the variable area nozzle that was evaluated during the comparative nozzle spray test. The severe hot spots in the upper right corners of the burner can exit after 5 hours and after 5.75 hours elapsed can be attributed to streaks in the nozzle spray pattern. (Heavy streaks were present in the spray of the variable area nozzle several times during the comparative nozzle spray test; consequently, this performance was expected.) At these times, the nozzles were plugged as indicated in figure 73 by the higher than normal inlet cluster pressure. The temperature maps obtained during run 5 indicate the severity of the effects of poor nozzle performance resulting from spraying contaminated fuel.

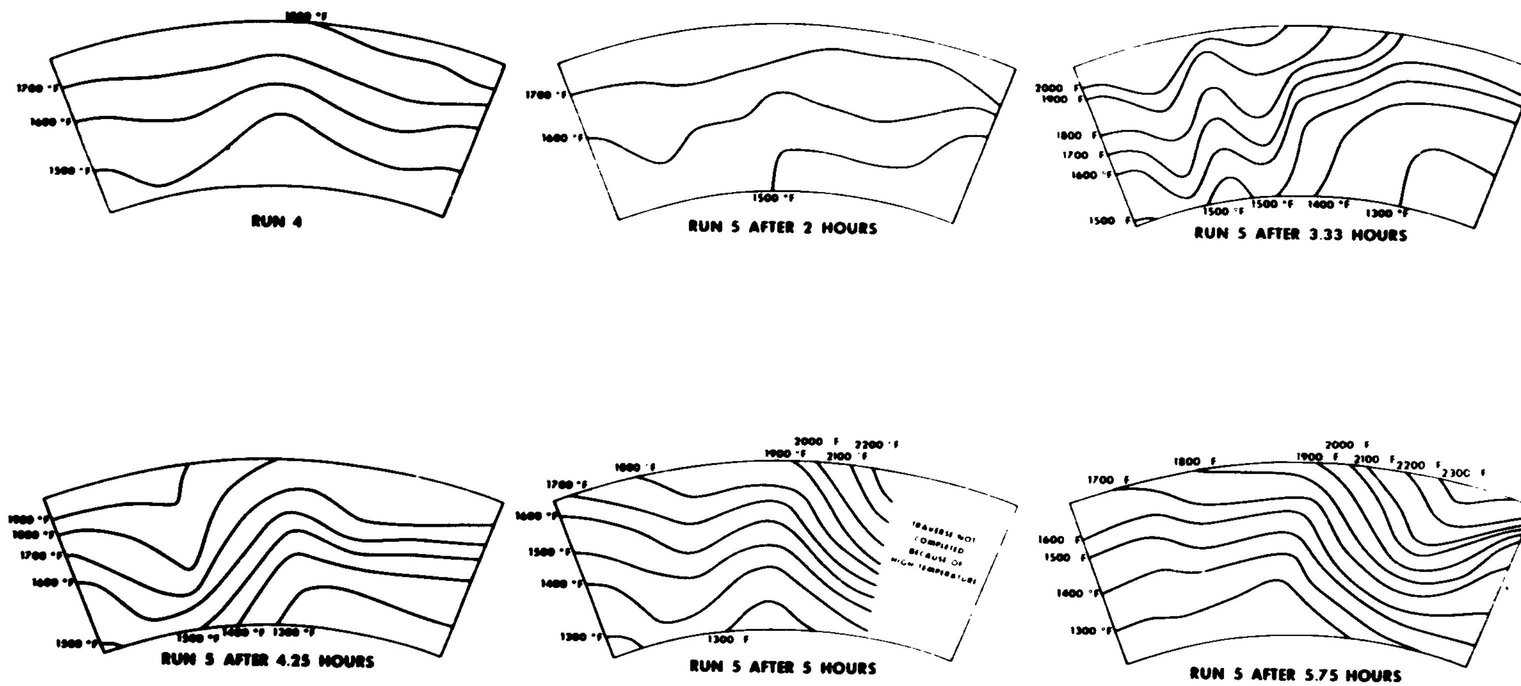
If the burner can temperature distributions that are presented for some of the contaminated fuel runs (especially run 5) occurred in an engine, the turbine blades or vanes would be damaged. Moreover, the damage would be more severe to the vanes than to the blades because the latter sense the average circumferential temperature, while the vanes sense the local temperature along their length. To assess the effect of a distortion in burner can exit temperature the average temperature profiles were computed and compared with similar profiles that were obtained in the base data runs with clean fuel. The results of this procedure are shown in figures 74 to 79. It should be noted that the profiles presented are for the most skewed temperature maps that were obtained during each contaminated fuel run. If the profiles obtained during the base data runs are assumed to be the profiles that allow the maximum turbine inlet temperature for a certain engine life, then the profiles shown for the contaminated fuel runs could result in a shortened vane life.



1. DATA FOR VARIABLE AREA DUAL ORIFICE NOZZLES
2. AVERAGE EXIT TEMPERATURE = 1600°F

Figure 66 Comparison of Burner Can Exit Temperature Maps – Runs 2 and 3 with 1

FD 352?



1. DATA FOR VARIABLE AREA NOZZLES

2. AVERAGE EXIT TEMPERATURE = 1600 °F

Figure 67 Comparison of Burner Can Exit Temperature Maps— Run 5 with 4

FD 3174A

FD 3521

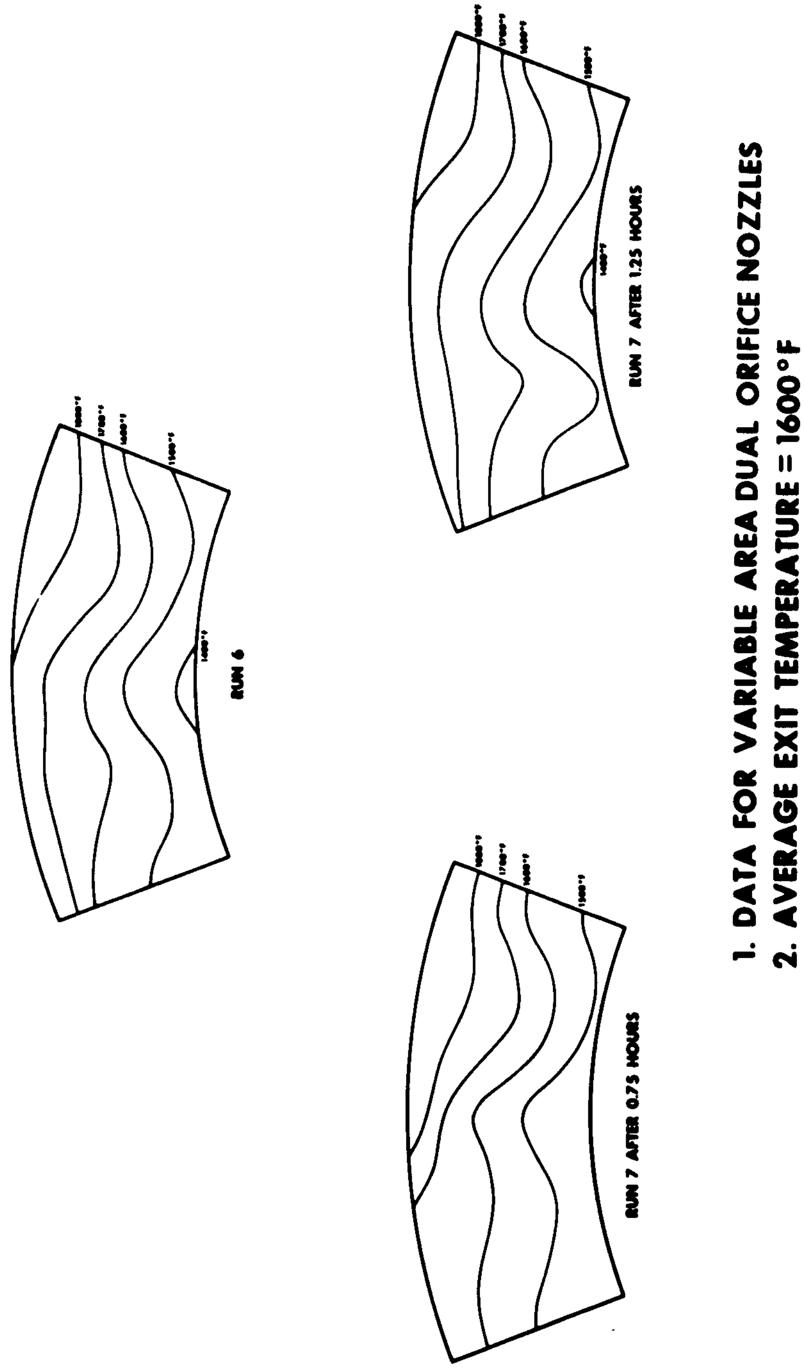
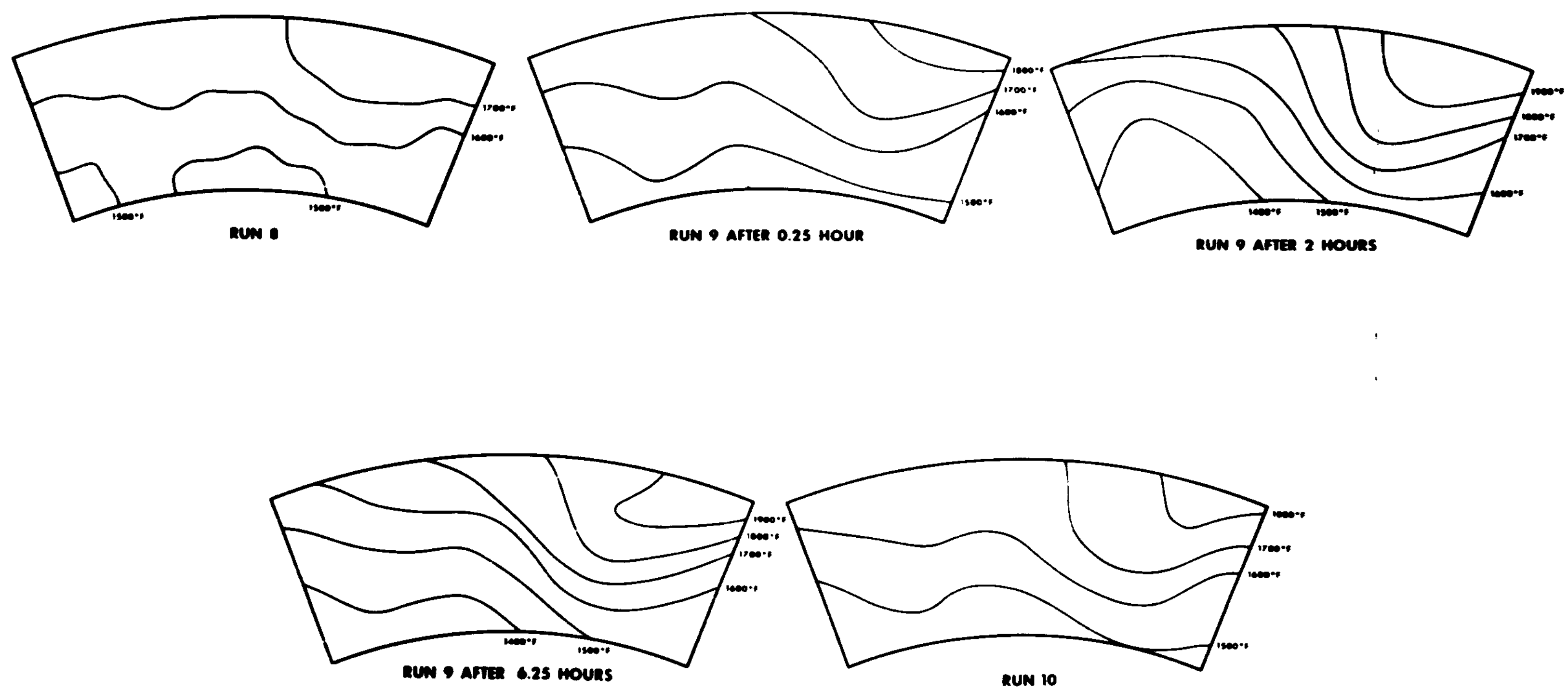


Figure 68 Comparison of Burner Can Exit Temperature Maps—Run 7 with 6



1. DATA FOR VARIABLE AREA DUAL ORIFICE NOZZLES

2. AVERAGE EXIT TEMPERATURE=1600 °F

Figure 69 Comparison of Burner Can Exit Temperature Maps— Runs 9 and 10 with 8

FD 3525

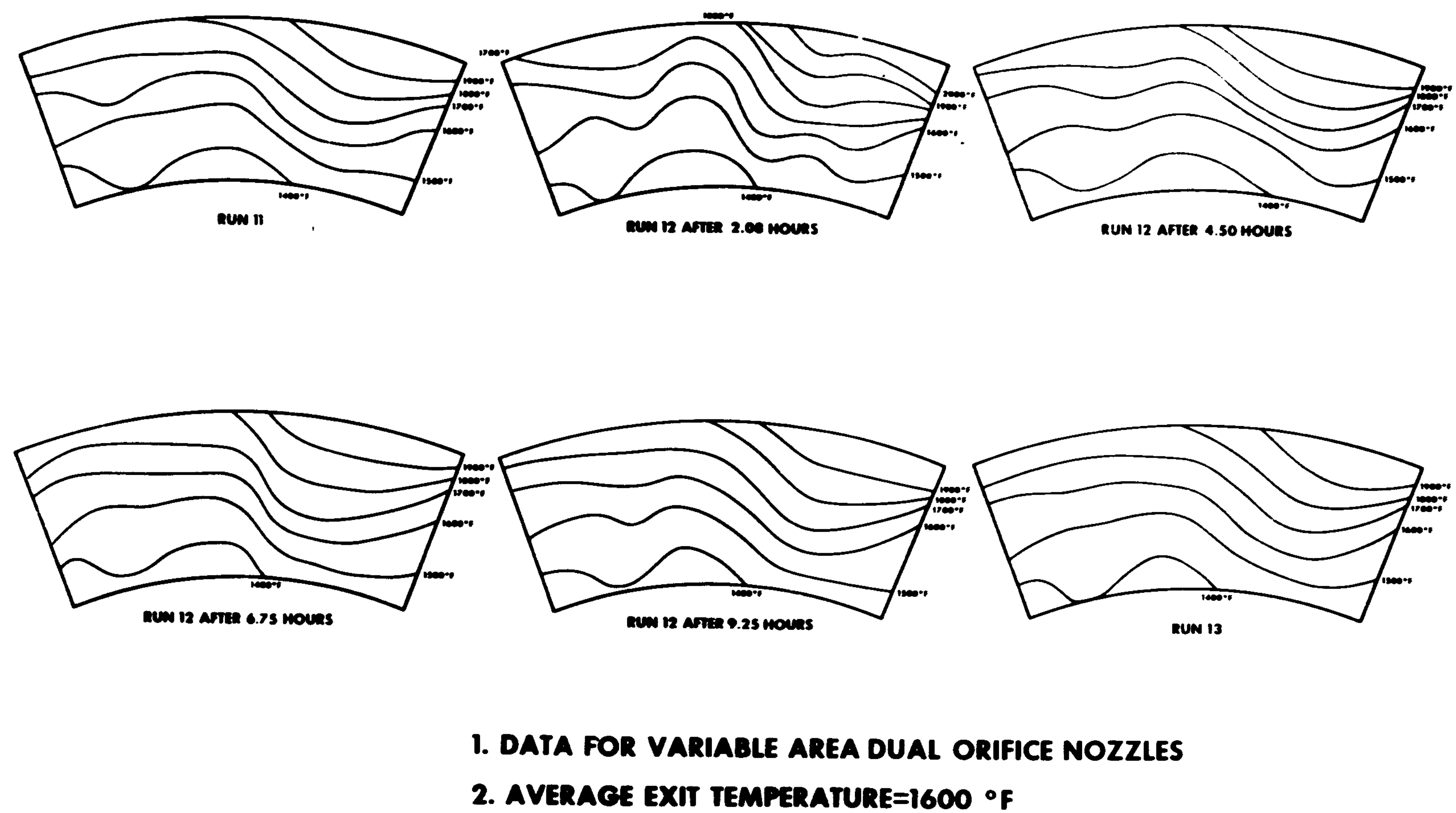
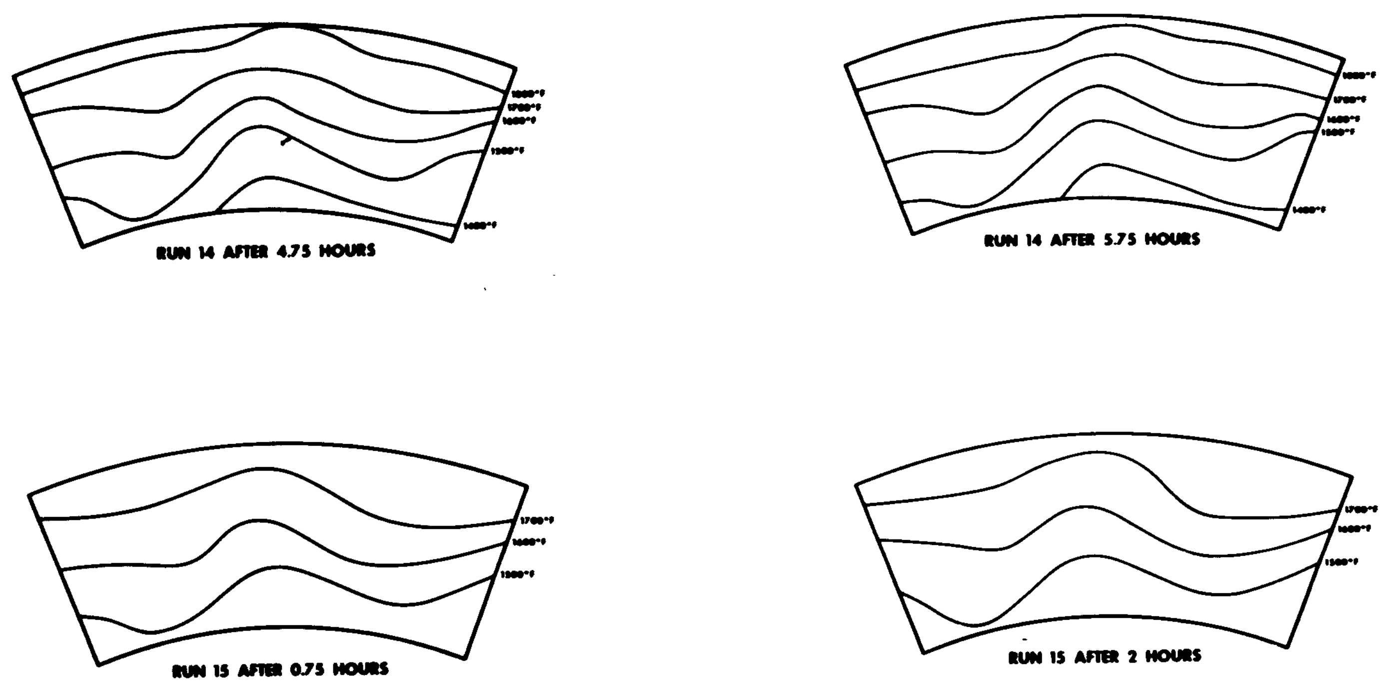


Figure 70 Comparison of Burner Can Exit Temperature Maps— Runs 12 and 13 with 11

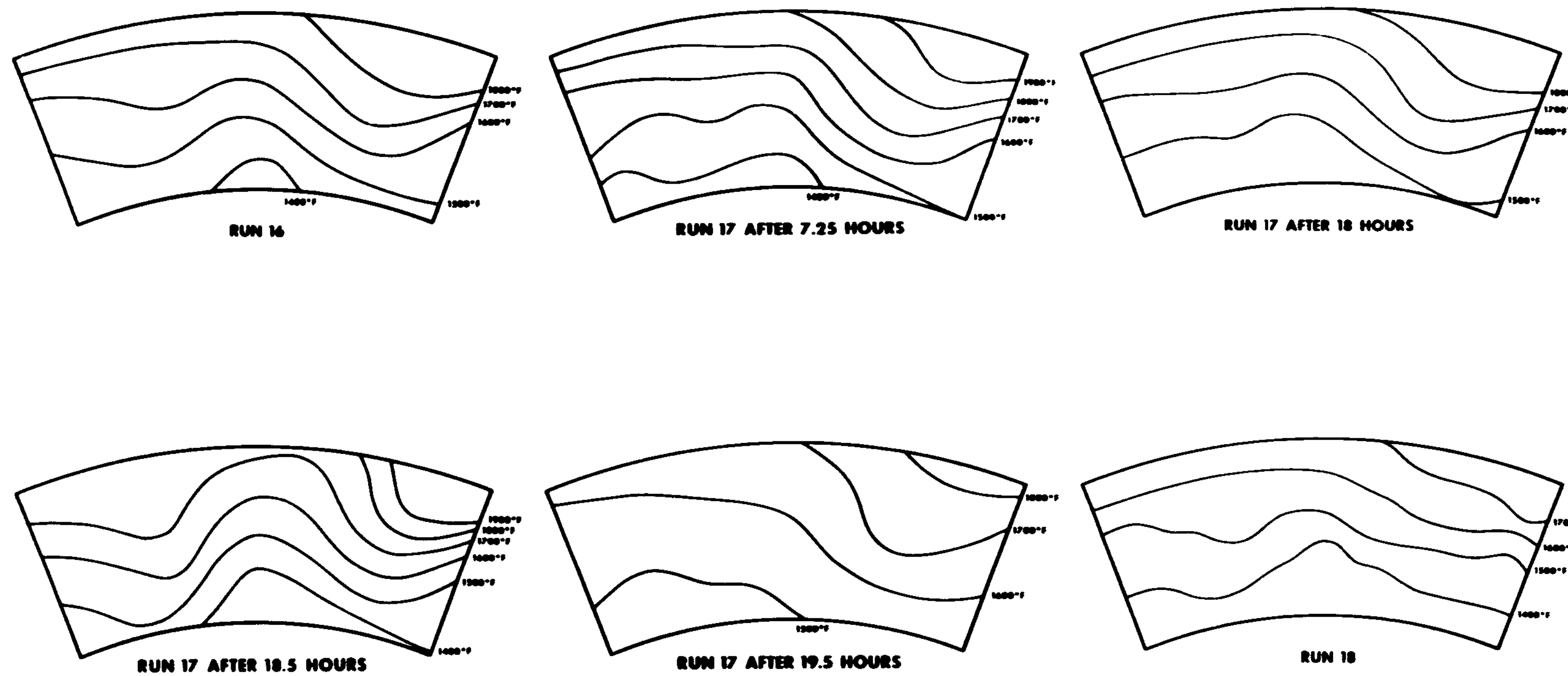
FD 3518



1. DATA FOR VARIABLE AREA DUAL ORIFICE NOZZLES
2. AVERAGE EXIT TEMPERATURE = 1600°F

FD 3520

Figure 71 Burner Can Exit Temperature Maps—Runs 14 and 15



1. DATA FOR VARIABLE AREA NOZZLES

2. AVERAGE EXIT TEMPERATURE = 1600°F

Figure 72 Comparison of Burner Can Exit Temperature Maps—Runs 17 and 18 with 16

FD 3550

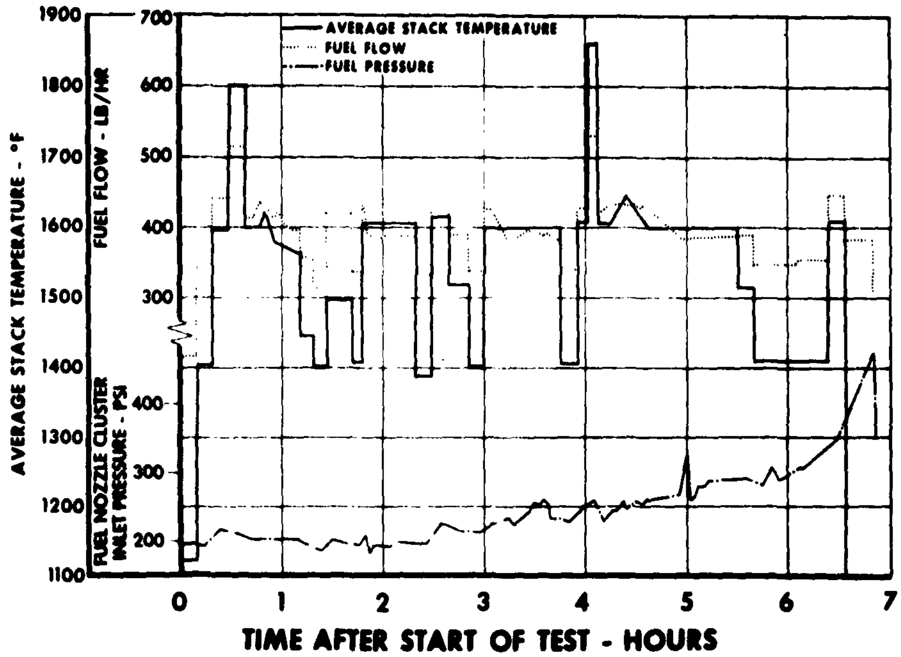


Figure 73 Test Condition Variations during Run No. 5 **FD 3569**

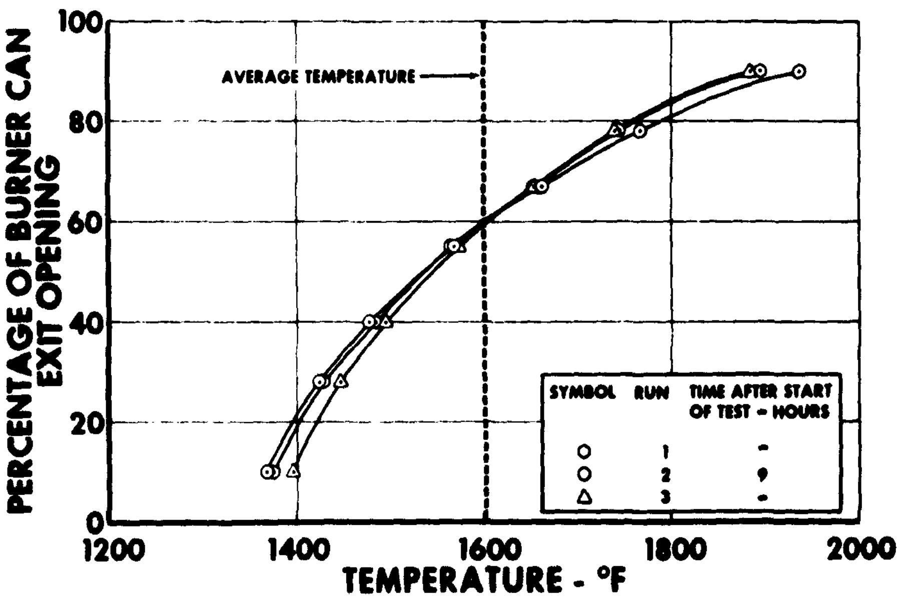


Figure 74 Burner Can Exit Temperature Profile Comparison—Runs 2 and 3 with 1 **FD 3567**

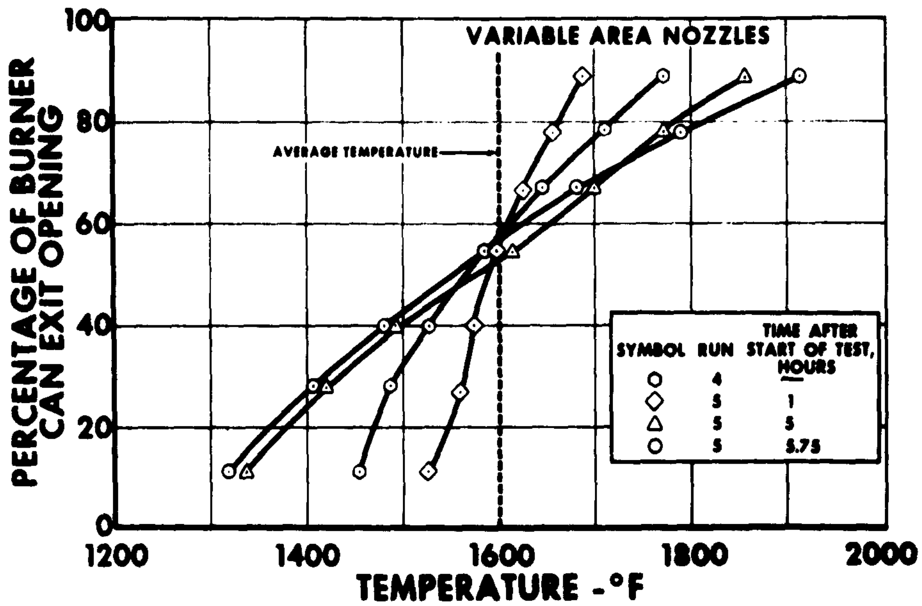


Figure 75 Burner Can Exit Temperature Profile Comparison— Run 5 with 4 FD 3546

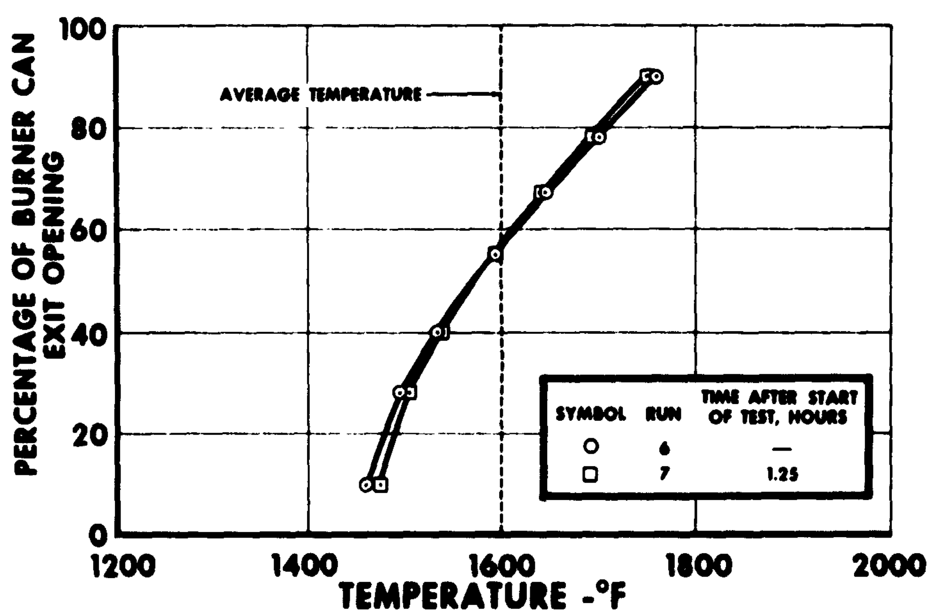


Figure 76 Burner Can Exit Temperature Profile Comparison— Run 7 with 6 FD 3490

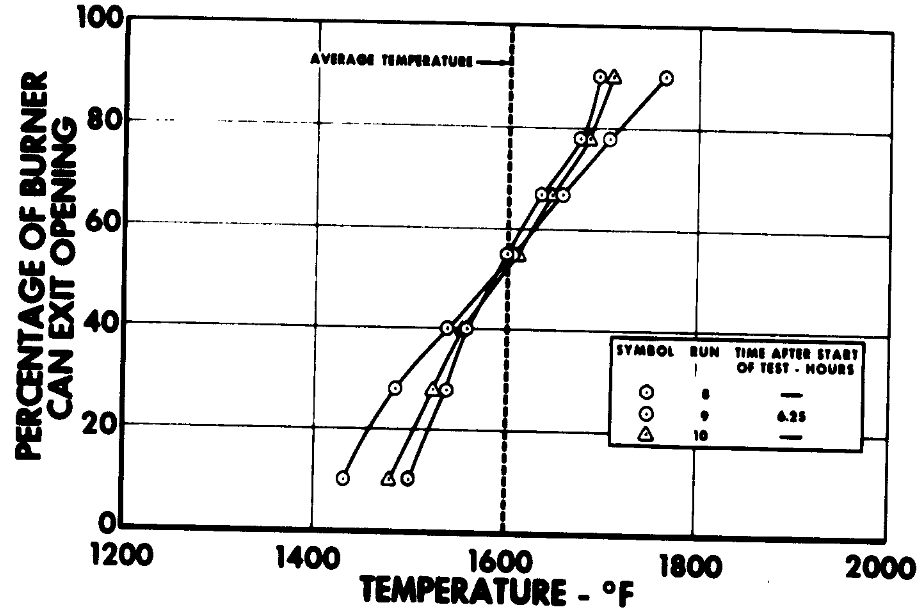


Figure 77 Burner Can Exit Temperature Profile Comparison— Runs 9 and 10 with 8 FD 3530

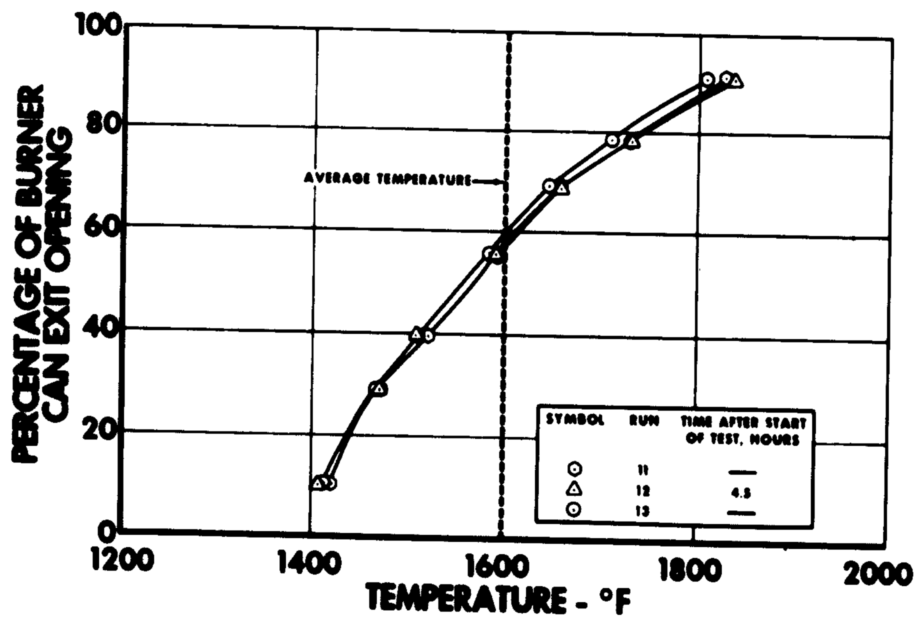


Figure 78 Burner Can Exit Temperature Profile Comparison— Runs 12 and 13 with 11 FD 3531

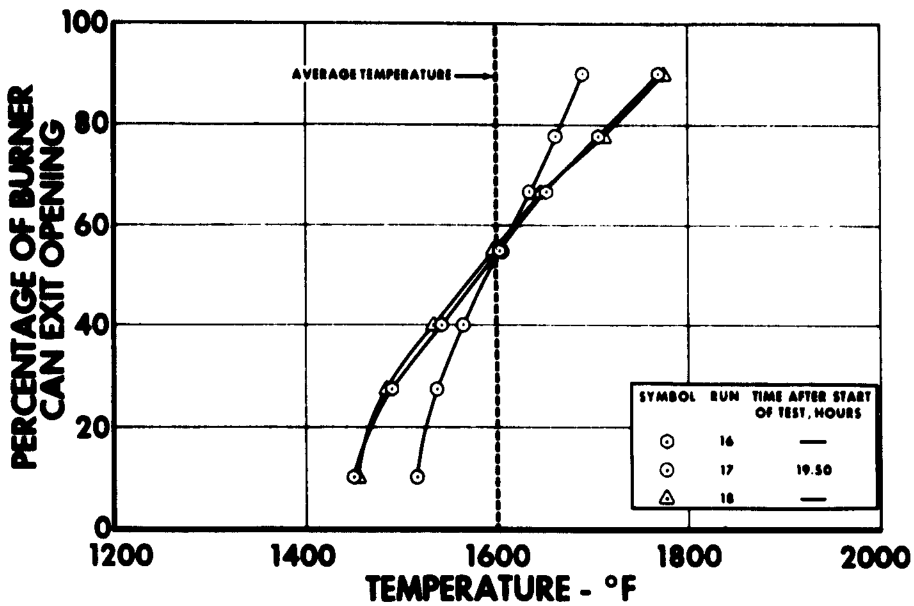


Figure 79 Burner Can Exit Temperature Profile Comparison— Runs 17 and 18 with 16 *FD 3548*

In Section VII the maximum positive deviations of the profiles for the contaminated fuel runs from the profiles obtained during the base data run were used to determine the effect on turbine vane life. These deviations for each contaminated fuel run are presented in table XI. The values presented for each of the contaminated fuel runs were obtained by comparing the profiles shown in figures 74 to 79 and selecting the maximum difference between the contaminated fuel run and its base data run.

TABLE XI. MAXIMUM DEVIATIONS OF CONTAMINATED FUEL RUN TEMPERATURE PROFILES FROM BASE DATA RUN TEMPERATURE PROFILES

Run No.	Maximum Deviation from Base Data Runs, °F
2	42
3	22
5	140
8	17
9	68
10	13
12	10
13	0
17	67
18	1

Figure 71 shows the maps taken during the calibration test of the cluster that was conditioned with the 60-hour cold-flow test with contaminated fuel. The maximum variation in temperature at the exit of the burner can that occurred during these runs was 455°F (obtained during run 14). This is less than the average variation (684°F) that was obtained during the runs with clean fuel.

c. Critical Burner Can Temperature Rise

The data for critical burner can temperature rise are presented in table X. These data agree in general with the data obtained with the traverse rake in that critical or allowable burner can temperature rise is decreased with contaminated fuel. Run 5 is again the run when the most severe reduction occurred.

The effect of contaminant accumulation on the critical burner can temperature rise is illustrated by the curve shown in figure 80. This curve shows that the critical burner can temperature rise dropped from 1400° to 700°F during run 5.

The critical burner can temperature rise, as defined in another portion of this section, is the burner can temperature rise at which the flame reaches the exit of the burner can. In these tests the critical temperature rise was lower than normal because uneven burning occurred in the burner can. Consequently, critical burner can temperature rise is a measure of the distortion of the burner can exit temperature distribution. Furthermore, because the temperature distributions at the burner can exit were determined, the critical burner can temperature rise data that were obtained during the tests were not used to determine the effect on turbojet performance in Section VII.

d. Burner Can Wall Temperatures

The envelope defined by the maximum and minimum values of burner can wall temperatures is shown in figure 81. Because all of the values are considerably below the melting point of the burner can material, the effect of contaminated fuel on burner can life is not considered to be significant, at least for the burner can evaluated. It should be pointed out that even when severe hot spots occurred at the exit of the burner can, none were detected on the burner can wall.

Only one burner can was used in this program. Following each run, and at the conclusion of the program, the can was inspected for holes in the wall. The burner can, which remained intact throughout the 18 runs, is shown bisected in figure 82.

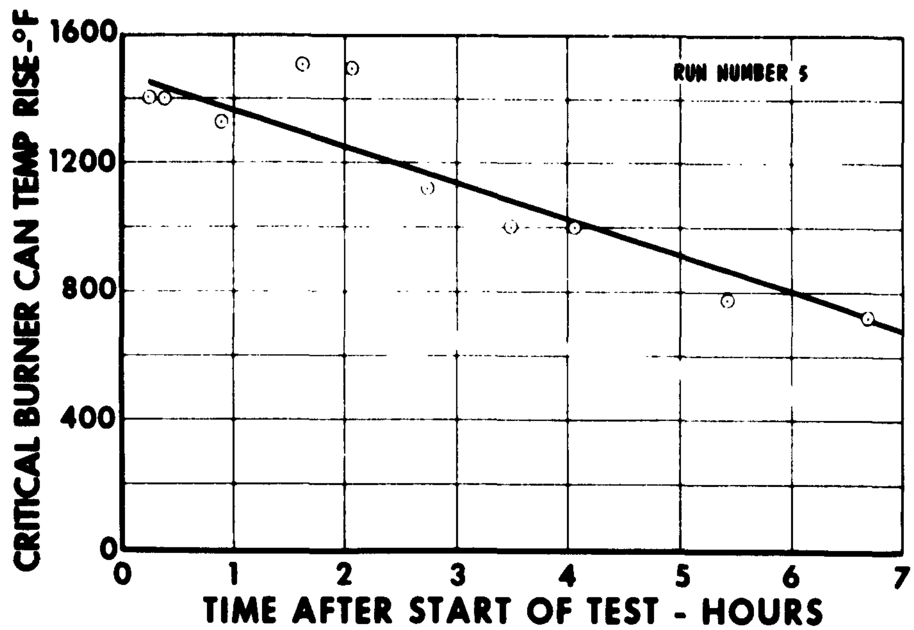


Figure 80 Effect of Contaminated Fuel on Critical Burner Can Temperature Rise—Run 5 FD 3507

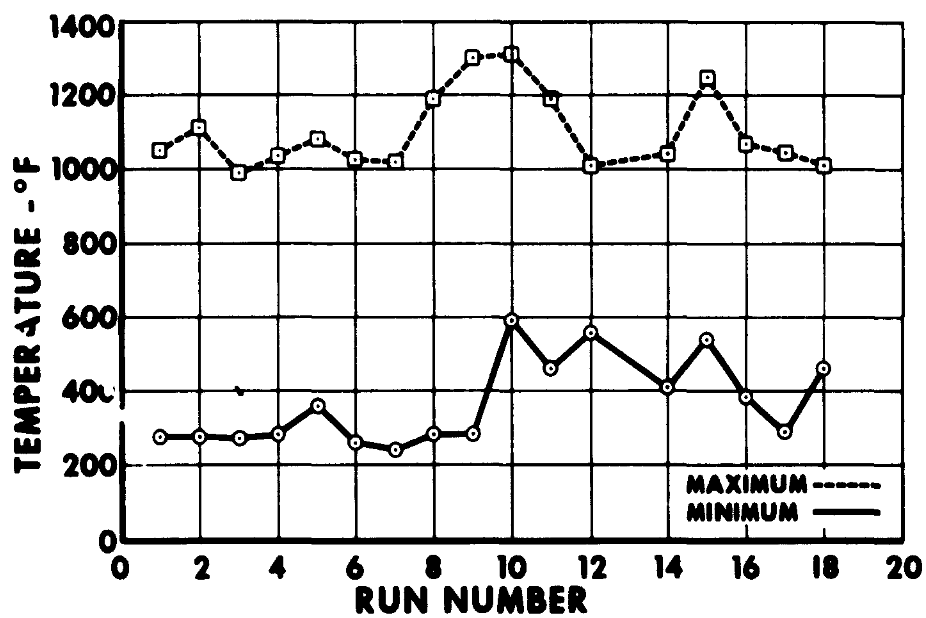


Figure 81 Variation of Burner Can Wall Temperature FD 3516

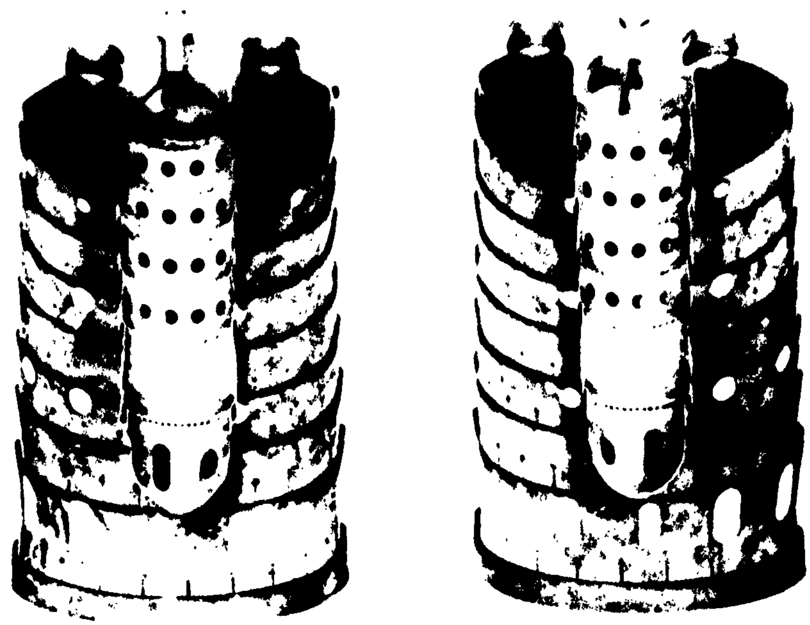


Figure 82 Bisected Burner Can **FE 19277**

e. Lean Blowout

The fuel flow at lean blowout varied from 10 to 100 lb/hr during the tests, as shown in table X. The corresponding values obtained with clean fuel are 50 to 60 lb/hr. The highest fuel flow at lean blowout occurred during run 15, which was the run in which the conditioned fuel cluster was used. This increase was not considered to be serious.

f. Pressure Loss

As can be seen in table X, the burner can pressure loss changed less than 2 psi during the burner can tests. This performance was expected because the burner can remained intact for the entire program.

SECTION VII

SPECIFIC STUDY OF THE EFFECTS OF CONTAMINATED FUEL ON TURBOJET PERFORMANCE

A. GENERAL

This section reports the results of a study in which the burner can test data were used to compute turbojet performance. The combustion efficiency data were used to compute the thrust specific fuel consumption at two altitude conditions and for both an afterburning and a nonafterburning engine. The burner can exit temperature profiles presented in Section VII were used to determine the effect of a distorted temperature profile at the burner can exit on turbine vane life. In addition, a computation was made in which it was assumed that the distortion in burner can exit temperature would be detected by the thermocouples in the inlet guide vanes. In this case the turbine inlet temperature could be reduced to a safe value; this procedure would result in a reduction in thrust and a change in thrust specific fuel consumption (TSFC).

The results of these computations are presented in two sections: effect of a reduction in combustion efficiency, and effects of a distorted temperature pattern at the burner can exit.

B. CONCLUSIONS

1. The maximum reduction of 8 percent in combustion efficiency obtained during run 9 of the burner can test resulted in an increase in TSFC of 14 percent for a nonafterburning engine. Although this increase is significant, the effect of contaminated fuel on TSFC is not considered to be serious because the combustion efficiency returned to a normal level almost immediately during the burner can test and because momentary increases in TSFC are not detrimental.

2. In an engine, hot spots at the burner can exit as severe as those that occurred during run 5 of the burner can tests will shorten the life of the turbine inlet guide vanes, although it may not affect turbine blade life. It has been estimated that the vane would fail in approximately 25 hours.

3. If the temperature distortions that occurred during run 5 were detected by the thermocouples in the inlet guide vanes, and if the turbine inlet temperature were reduced to a safe value, the net thrust would be reduced 9.4 percent.

C. DISCUSSION

1. *EFFECT OF A REDUCTION IN COMBUSTION EFFICIENCY*

By use of the combustion efficiency data obtained during the burner can test, the effects on TSFC were determined for a sea level condition and an altitude condition. In these computations, the curves of TSFC variation with combustion efficiency presented in figures 83 and 84 were used:

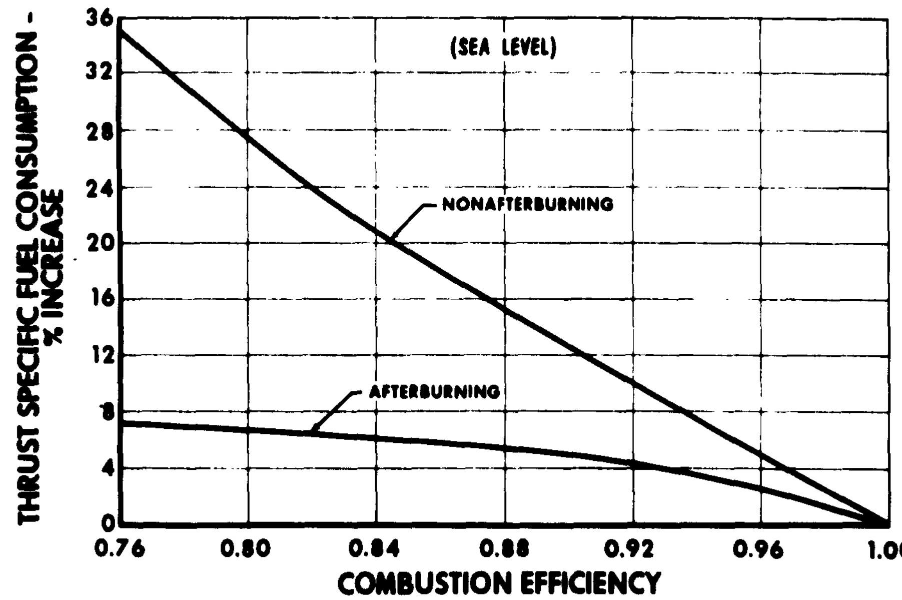


Figure 83 Thrust Specific Fuel Consumption Variation with Combustion Efficiency (Sea Level) *FD 3389*

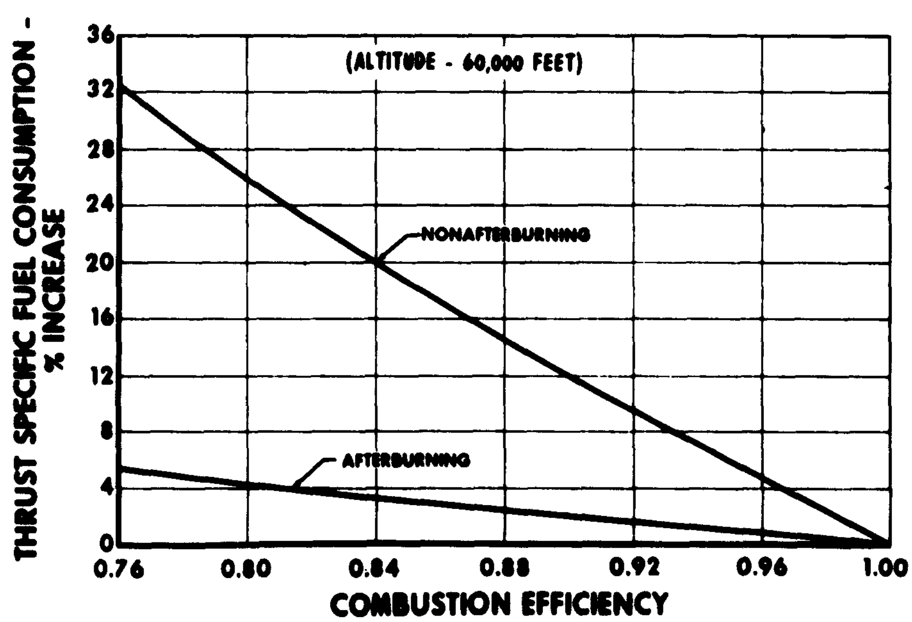


Figure 84 Thrust Specific Fuel Consumption Variation with Combustion Efficiency (60,000 ft) *FD 3390*

these curves are considered representative of current turbojets. The maximum reduction in combustion efficiency for each of the contaminated fuel runs is presented in figure 65. Use of these values results in the changes in TSFC that are presented in table XII for the cruise conditions specified. (The changes in TSFC for the first four contaminated fuel runs are not presented because the combustion efficiencies for these runs were above the base data.) Because combustion efficiency did not remain at a low level for an extended period of time, these increases in TSFC are not considered to be serious.

TABLE XII. EFFECT OF CONTAMINATED FUEL ON TSFC

Run No.	TSFC Increases, Percent			
	Sea Level		60,000 Ft.	
	Non A/B	A/B	Non A/B	A/B
9	14.1	1.45	12.5	2.1
10	0.7	0.05	0.6	0.05
12	4.7	0.75	4.6	0.70
13	3.3	0.50	3.2	0.50
17	8.1	0.90	7.3	1.1
18	2.8	0.30	2.5	0.4

2. *EFFECTS OF A DISTORTED TEMPERATURE PATTERN AT THE BURNER CAN EXIT*

a. *Effect on Vane Life*

The life of a turbine blade or a vane is the time required for it to creep to a length where further use would cause engine failure. This life is a function of the loads and temperatures to which the blades or vanes are subjected. A higher than normal temperature at the turbine inlet can shorten the life of the blades and vanes.

As explained in Section VI, maximum deviations of the contaminated fuel run profiles from the base data runs are to be used to determine the effect on turbine vane life. (Blade life should not be severely affected unless the temperature at some location approaches the blade material's melting point, because the blades sense the average circumferential temperature at each radial location.) Assuming that (1) the profiles obtained during the base data runs are ideal and (2) the vanes have a life of 3000 hours (the time required for the vane to creep 1%) with the base data run profiles, life will be shortened, if the temperature profile deviates from this profile. Using the maximum deviations presented in table XII, the corresponding vane lives for the profiles obtained during each contaminated fuel run is presented in table XIII. It should be noted that the hours given for the expected vane life are considered to be pessimistic because the values presented in the table are the vane lives that could be realized if the profiles did not change. For deviations from the ideal that occur for short periods of time, the vane life would

not be significantly affected. The procedure used in obtaining these values also results in pessimistic values since the maximum deviation was used and, therefore, only at the point where this condition occurs would the vane creep 1 percent in the time presented. At other locations along the vane, the creep rate would be less and, consequently, the creep rate for the vane length would be less than 1 percent.

TABLE XIII. EFFECT OF CONTAMINATED FUEL ON VANE LIFE

Run No.	Maximum Deviation from Base Data Run, °F	Expected Life, Hours
2	42	500
3	22	1150
5	140	25
8	17	1400
9	68	185
10	13	1700
12	10	1900
13	0	3000
17	67	2000
18	1	2800

Even though the values presented in table XIII may be pessimistic, it should be noted that only the results of run 5 would cause failure before 60 hours (the time that has been considered to be desirable engine operating time with contaminated fuel). This run was made with the variable area nozzles; all of the other runs were made with the variable area dual orifice nozzles. Based on this consideration, the variable area dual orifice nozzles again appear to be satisfactory for operation with contaminated fuel.

b. Effect on Engine Thrust and TSFC

If the distorted profiles are detected by the thermocouples in the inlet guide vanes, the fuel flow could be reduced to an allowable value. This procedure would result in a reduction in net thrust and a change in TSFC.

Using the values presented in table XII, the resulting reductions in thrust and TSFC can be computed for each contaminated fuel run. The results of these computations are presented in table XIV. The curves for thrust and TSFC versus burner can temperature rise presented in figures 85, 86, 87, and 88 were used in making these computations.

TABLE XIV. EFFECT OF CONTAMINATED FUEL ON THRUST AND TSFC

SEA LEVEL

Run No.	Max. Deviation From Base Data Run	Burner Can Temperature Rise	Non A/B Change In TSFC, %	Non A/B Change In Net Thrust, %	A/B Change In TSFC, %	A/B Change In Net Thrust, %
2	42	1326	−0.75	−3.1	+2.2	−1.2
3	22	1346	−0.40	−1.2	+1.2	−0.6
5	140	1228	−2.00	−9.4	+7.7	−4.5
8	17	1351	−0.39	−1.1	+0.90	−0.4
9	68	1300	−1.1	−4.7	+3.7	−2.0
10	13	1355	−0.3	−0.80	+0.7	−0.4
12	10	1358	−0.2	−0.5	+0.5	−0.2
13	0	1368	0.0	0.0	0.0	0.0
17	67	1301	1.1	−4.6	+3.6	−2.0
18	1	1367	−0.01	0.0	+0.01	0.0

60,000 FT.

Run No.	Max. Deviation From Base Data Run	Burner Can Temperature Rise	Non A/B Change In TSFC, %	Non A/B Change In Net Thrust, %	A/B Change In TSFC, %	A/B Change In Net Thrust, %
2	42	1326	−0.90	−2.5	+1.2	−1.1
3	22	1346	−0.50	−1.5	+0.6	−0.5
5	140	1228	−2.0	−8.9	+4.4	−3.9
8	17	1351	−0.45	−0.9	+0.4	−0.4
9	68	1300	−1.30	−4.2	+2.0	−1.7
10	13	1355	−0.40	−0.60	+0.3	−0.3
12	10	1358	−0.30	−0.45	+0.2	−0.2
13	0	1368	0.0	0.0	0.0	0.0
17	67	1301	−1.25	−4.1	+2.0	−1.7
18	1	1367	−0.01	0.0	0.0	0.0

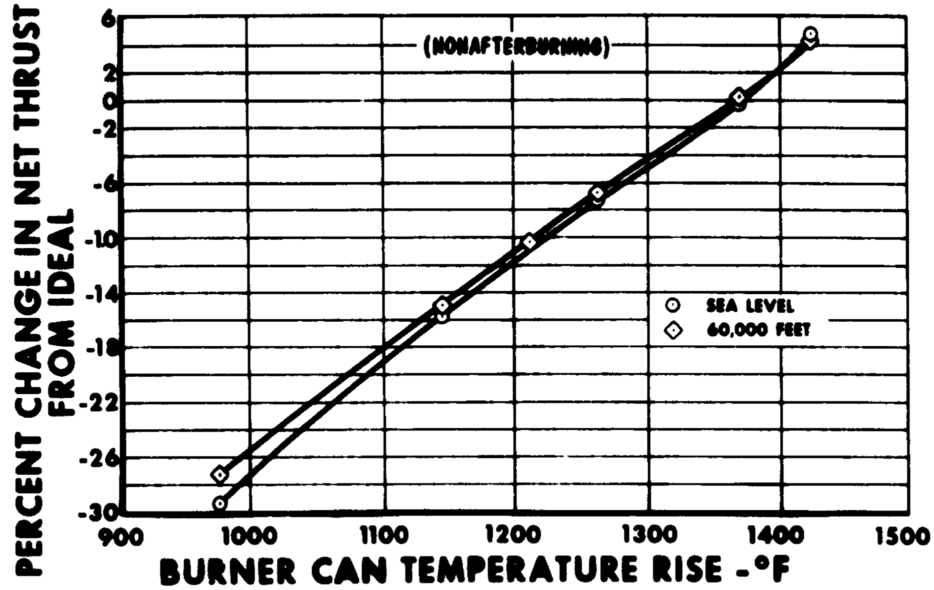


Figure 85 Change in Net Thrust with Burner Can Temperature Rise (nonafterburning) *FD 3506*

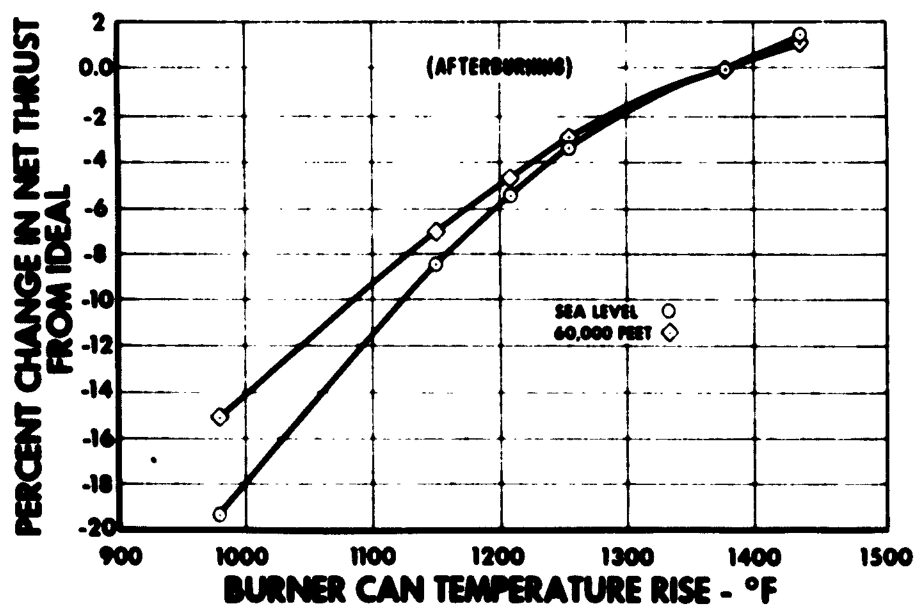


Figure 86 Change in Net Thrust with Burner Can Temperature Rise (afterburning) *FD 3391*

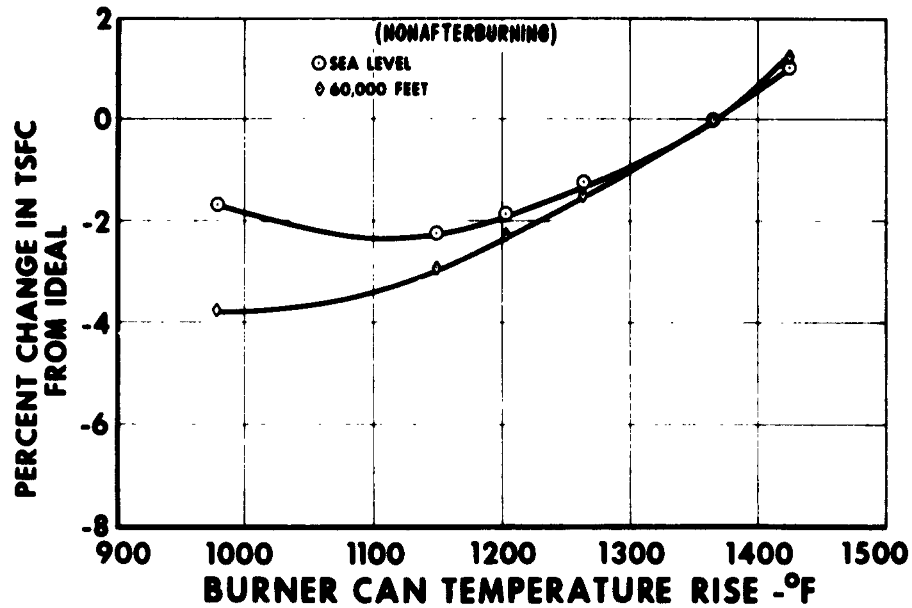


Figure 87 Thrust Specific Fuel Consumption vs Burner Can Temperature Rise (nonafterburning) *FD 3393*

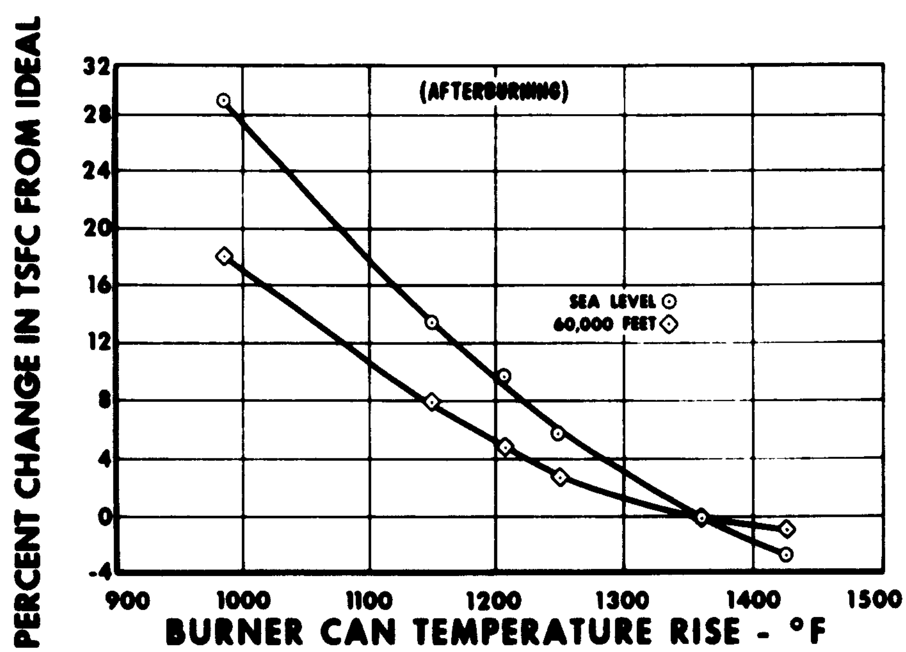


Figure 88 Thrust Specific Fuel Consumption vs Burner Can Temperature Rise (afterburning) *FD 3394*

SECTION VIII

ANALYTICAL NOZZLE DESIGN STUDY

A. GENERAL

An analytical nozzle design study was required under Work Statement Item 3. (Delavan Manufacturing Company was subcontracted to conduct a portion of this study.) The objective of the analytical nozzle design study was to determine the best method of atomizing contaminated fuel. In the study, three types of atomizers were considered: pressure, air, and impingement. The specifications in table XV were used for the study.

TABLE XV

NOZZLE SPECIFICATIONS FOR ANALYTICAL NOZZLE DESIGN STUDY

Nozzle Inlet Pressure, psig	Fuel Flow, lb/hr	Spray Cone Angle Limits, degrees
150	34	75 to 85
–	894	75 to 85

NOTE: The fuel flows given are for a test fluid that is maintained at a temperature of 80° ± 12°F, and that has the following characteristics at this temperature:

Specific Gravity = 0.768
Kinematic Viscosity = 1.8 centistokes

In addition, it was assumed that the mean droplet size must be 150 microns or less for good atomization. These specifications were used in designing the variable area dual orifice nozzle (Nozzle No. 4) of the comparative nozzle spray test and the variable area nozzle (Nozzle No. 5). These specifications were selected for the study for two reasons: (1) the performance of the Nozzles No. 4 and No. 5 during the comparative nozzle spray tests and the burner can tests were available for background information and (2) the specified pressure at the low flow was above that required to achieve acceptable atomization.

The pressure requirement at the low fuel flow arises because the fuel is used for cooling auxiliary equipment in the aircraft. As a result, because the fuel is at a high temperature in the fuel system, it could be vaporized if a high fuel pressure were not maintained. This pressure requirement results in nozzles that have smaller metering passages than those in a nozzle designed to give acceptable atomization (a spray with a mean droplet size of 150 microns).

The utilization of this pressure specification in the study illustrates that there are other nozzle requirements rather than just good atomization. Because of these additional requirements, some nozzle types are not acceptable.

For the flow specifications presented in table XIII the analytical nozzle study was made of the three type nozzles: pressure atomizing, air atomizing, and impingement. For each of these types, the nozzles are compared in the following paragraphs on the basis of the following characteristics:

1. Adaptability for flow schedule
2. Passage size
3. Energy requirements

B. CONCLUSIONS

1. For a given flow range, the air atomizing nozzle has the largest metering passages. Consequently, if atomization is the only nozzle requirement, air atomizing nozzles would be best for spraying contaminated fuel.

2. If in addition to atomization, there are other nozzle requirements such as a minimum nozzle pressure drop, the conventional dual orifice nozzle (with a flow divider valve for scheduling flow to the secondary orifice) is a good nozzle choice for spraying contaminated fuel.

   The dual orifice nozzle system is the best nozzle for spraying contaminated fuel in the flow range from 34 to 890 lb/hr when a minimum pressure of 150 psig must be maintained upstream of the nozzle.

3. The metering valve that is required in the return line of the spill nozzle makes the desirability of this nozzle questionable.

4. The orifice size of an impingement nozzle is the smallest of the nozzles considered. Consequently this nozzle type is least desirable for use with contaminated fuel.

C. DISCUSSION

1. PRESSURE ATOMIZING NOZZLES

a. Background

Because of the results of the comparative nozzle spray tests a spill nozzle and a dual orifice nozzle were designed in this study. Both of these nozzles are swirl type nozzles; the important dimensions of this nozzle type are:

(1) Outlet orifice diameter

(2) Swirl chamber diameter

(3) Swirl slots — number and size

(4) The angle made by the slots with the axis of the orifice (normally this parameter is defined by the angle made by the slots with a line perpendicular to the axis of the nozzle. In the following pages this angle will be referred to by α.)

The dual orifice nozzle and spill nozzle are discussed separately in subsequent paragraphs.

b. The Dual Orifice Nozzle

A dual orifice nozzle with a flow divider valve can be designed to meet all of the flow specifications. It should be noted that this nozzle system is the same as the variable area dual orifice (VADO) nozzle that was used throughout this program. The only difference between the variable area dual orifice nozzle and the dual orifice nozzle with an integral flow divider valve which was designed in this study is the location of the flow divider valve. In the VADO nozzle, this valve is at the inlet to the swirl chamber of the secondary orifice; the valve varies the area of the swirl slots. In the analytical studies a standard dual orifice nozzle with the flow divider located upstream is designed. This latter arrangement might have an advantage in that malfunction of the valve will not cause a poor spray pattern. In the VADO nozzle, streaks might occur in the spray pattern if the poppet becomes stuck open widely. (It should be noted, however, that during the last nozzle spray test the VADO performed satisfactorily with contaminated fuel.)

The secondary orifice of the dual orifice nozzle was designed first. Since the primary orifice is inside the secondary orifice, the latter places a limit on the size of the primary orifice. It was assumed that this orifice must provide a spray with a mean droplet size for the flow range 240 to 890 lb/hr. This flow range was selected on the assumption that acceptable atomization will be achieved for the entire flow range (34 to 890 lb/hr) by the following scheme:

Primary and Secondary	34 to 240 lb/hr
Secondary only	240 to 890 lb/hr

Also the orifice spray angle must be between 75 and 85 degrees for this flow range. The secondary orifice was designed by the following procedure and by use of the equations presented in Appendix D.

(1) A pressure drop of 450 psi was assumed for the maximum flow rate of 890 lb/hr.

(2) A discharge coefficient was assumed. The orifice outlet diameter could then be determined for the conditions given in (1).

(3) The nozzle outlet diameter, the flow rate, the 80 degree spray angle requirement, and the assumed discharge coefficient permitted the determination of the other nozzle geometry.

(4) The droplet size was then determined for the flow rate of 240 lb/hr. If the droplet size was above 150 micron, the procedure was repeated for a different discharge coefficient.

The results of this procedure are shown in figure 89, where the droplet size for a flow rate of 240 lb/hr and the nozzle dimensions are plotted against the discharge coefficient. As shown, a mean droplet diameter of 150 microns requires that the discharge coefficient be 0.35. For this discharge coefficient the orifice dimensions can be obtained from figure 89.

These are:

Outlet Orifice Diameter	=	0.114 inch
Total Slot Area	=	0.02[illegible] in.^{2} (for $\alpha = 0$)
Swirl Chamber Diameter	=	0.37 inch

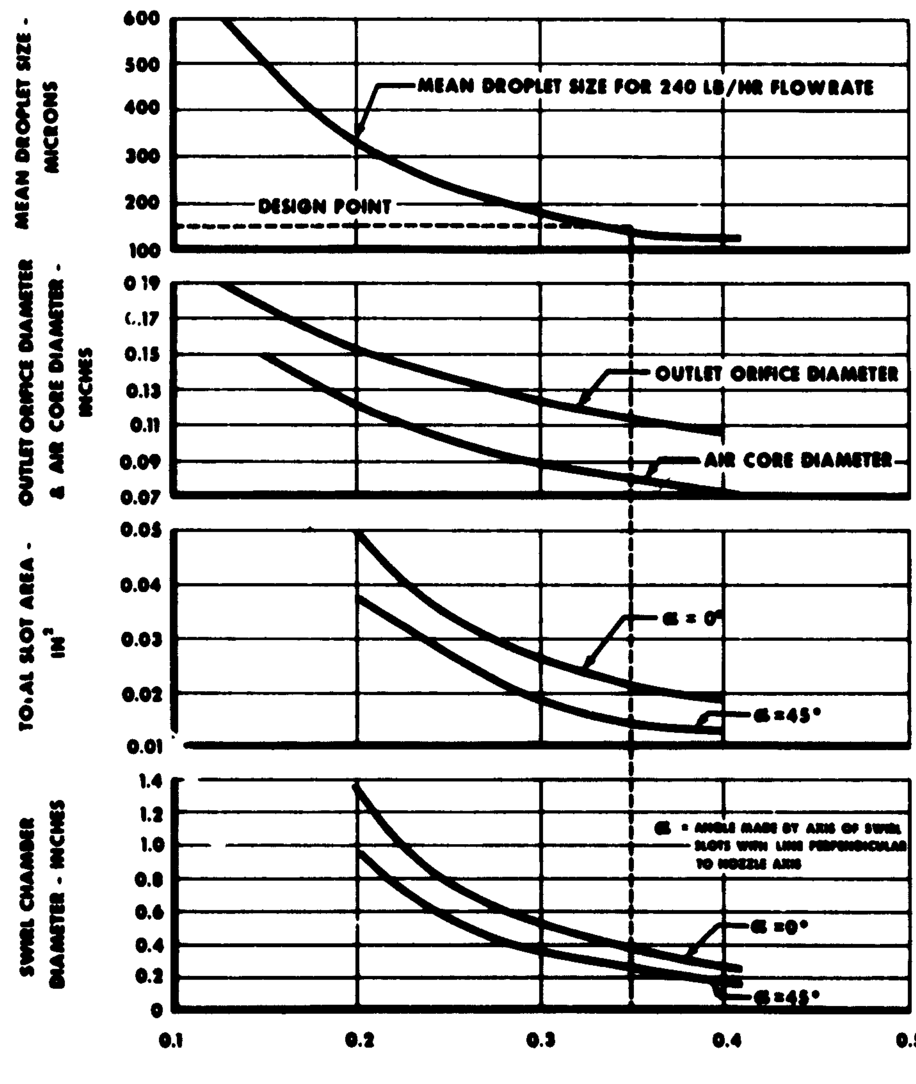


Figure 89 Design Curves for Secondary Orifice of the Dual Orifice Nozzle FD 3522

It should be noted that it was assumed that the swirl slots were perpendicular to the axis of the nozzle ($\alpha = 0$). This permits maximum dimensions as indicated by the curves for total slot area. In the secondary orifice this can be achieved because the orifice outside diameter is not as critical as in the primary orifice. It is important to note that values other than zero are necessary only when the nozzle envelope is critical.

The slot dimensions, of course, are a function of the number of slots used. The number of slots are selected on the basis of orifice performance. Delavan Manufacturing Company has found that a small number of slots give satisfactory performance in a small orifice, while a large number are required in large orifices. According to Delavan, eight slots are needed in the secondary orifice, and one slot can be used in the primary orifice. If eight slots were used in the secondary orifice, the slot width would be 0.053 inch.

With the secondary orifice sized, the primary orifice was then designed. The outside diameter of the portion of the primary orifice that extends into the outlet diameter of the secondary orifice must be less than the secondary orifice air core diameter; otherwise, the presence of the primary orifice would affect the performance of the secondary orifice adversely. The diameter of the air core in a swirl type nozzle is a function of the discharge coefficient and the outlet orifice diameter. The air core of the secondary orifice is plotted against the discharge coefficient in figure 89. As seen in the figure the air core diameter for the selected secondary orifice is 0.08 inch. If the primary orifice outside diameter were this size, the thickness of the annulus between the primary and the secondary orifices would be only 0.017 inch, which is considerably less than the secondary swirl slot dimensions even if as many as eight slots are used. In the design studies, the primary orifice was sized for several annulus thicknesses. A primary orifice design was selected that made the annulus as large as the primary orifice swirl slot. It should be noted that this represents a compromise between primary orifice size and annulus thickness. An annulus thickness of 0.03 inch was selected. For this thickness the outside diameter of the primary orifice is 0.054 inch. If it is assumed that a practical wall thickness at the tip is 0.01 inch for the primary orifice, then the corresponding primary outlet orifice is 0.034 inch.

To achieve the 150 psig minimum pressure requirement at 34 lb/hr fuel flow with an outlet orifice diameter of 0.034 inch, the discharge coefficient must be 0.25. The design point is shown in figure 90, where the dimension of the primary orifice and the droplet size variation are plotted against the orifice discharge coefficient. It should be noted the droplet size is much below the 150-micron requirement for all values of the discharge coefficient; this points out that the 150 psi pressure requirement is in excess of that required for good atomization.

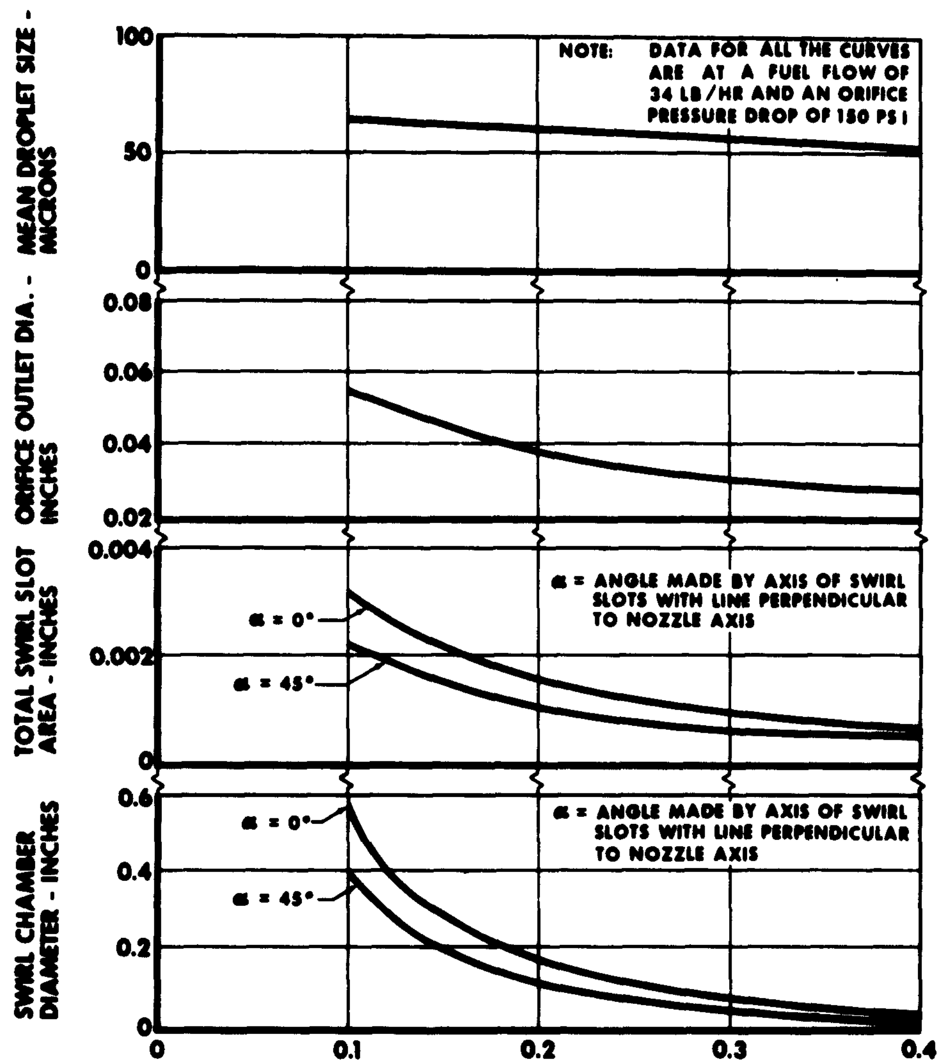


Figure 90 Design Curves for Primary Orifice of the Dual Orifice Nozzle *FD 3571*

The angle α was assumed to be 45 degrees. This is considered to be a practical value, because small values of α require large swirl chamber outside diameters and because the primary orifice size is limited by the secondary orifice. For this value, the total slot area is 0.008 square inch and the swirl chamber diameter is 0.07 inch. Both of these values can be obtained from figure 90. One slot will be used. (Delavan has found that one slot in the primary orifice of a dual orifice nozzle of this approximate size provides satisfactory spray uniformity.) The slot will be 0.0284 inch square. This value is approximately the same as the annulus thickness between the primary and the secondary orifices.

A summary of the dimensions of the dual orifice nozzle are presented in table XVI.

TABLE XVI. DUAL ORIFICE NOZZLE DIMENSIONS

Orifice	Outlet Orifice Diameter, inch	Swirl Slot Dimension, inch	Swirl Chamber Diameter, inch	Number of Slots
Primary	0.034	0.0284 x 0.0284	0.07	1
Secondary	0.114	0.053 x 0.053	0.37	8

c. The Flow Divider Valve

A flow divider valve is required to schedule the flow to the secondary orifice. The flow requirements of this valve are (1) 150 psig opening pressure, and (2) valve metering area to be sized for 890 lb/hr at a pressure drop of 200 psi. With a valve designed for these specifications the flow schedule for the dual orifice nozzle would be as shown in figure 91.

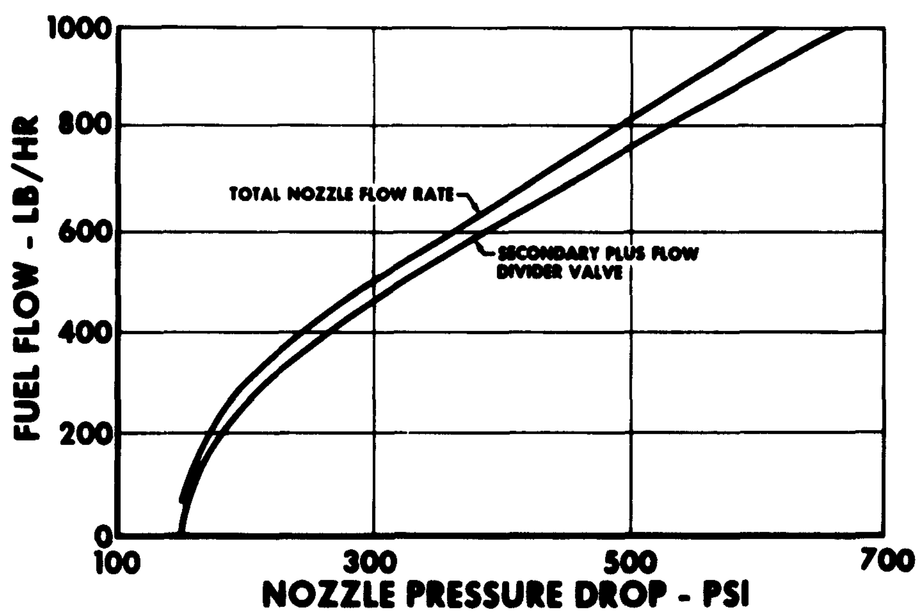


Figure 91 Flow Schedule, Dual Orifice Nozzle *FD 3535*

The flow divider valve requirements for both the dual orifice nozzle and the impingement nozzle are the same since the same flow schedule is being sought; therefore the design of the flow divider valve will be presented only once.

Delavan has designed at least one valve for use with fuel contaminated with MIL-E-5007A contaminant; their valve proved successful during a 50-hour test. Delavan's nozzle had the following important features:

(1) A metering area with large minimum openings

(2) Close clearances

(3) Hard materials

(4) Large shear forces.

To illustrate these features, two valves have been designed. These are shown in figures 92 and 93. The first valve is a conventional type that has an annular metering area; in the second valve slots are uncovered as the piston advances. In this latter arrangement the maximum opening for a given area is obtained by use of the slots. The valve specifications that were presented in the preceding section were used.

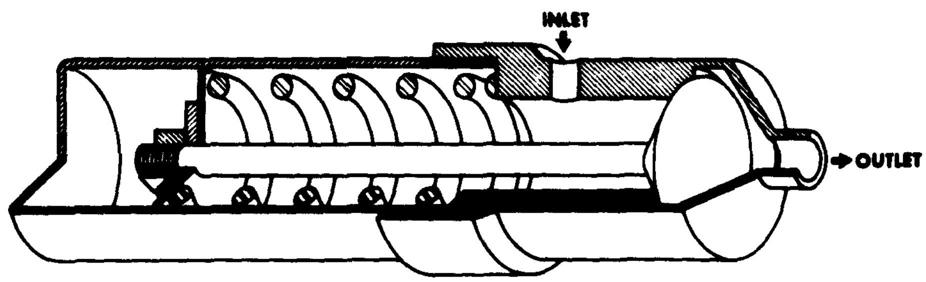


Figure 92 Typical Flow Divider Valve (Valve 1) *FD 3557*

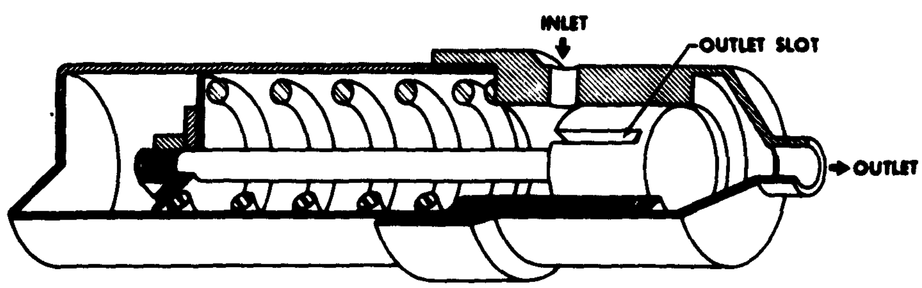


Figure 93 Contaminated Fuel Flow Divider Valve (Valve 2) *FD 3563*

With the foregoing specifications, valve 1 was designed by assuming a valve seat diameter of 3/16 inch. With this diameter selected the spring constant required is 0.00517 inch/lb_f and the pressure, area, and valve lift are functions of fuel flow as shown in figure 94.

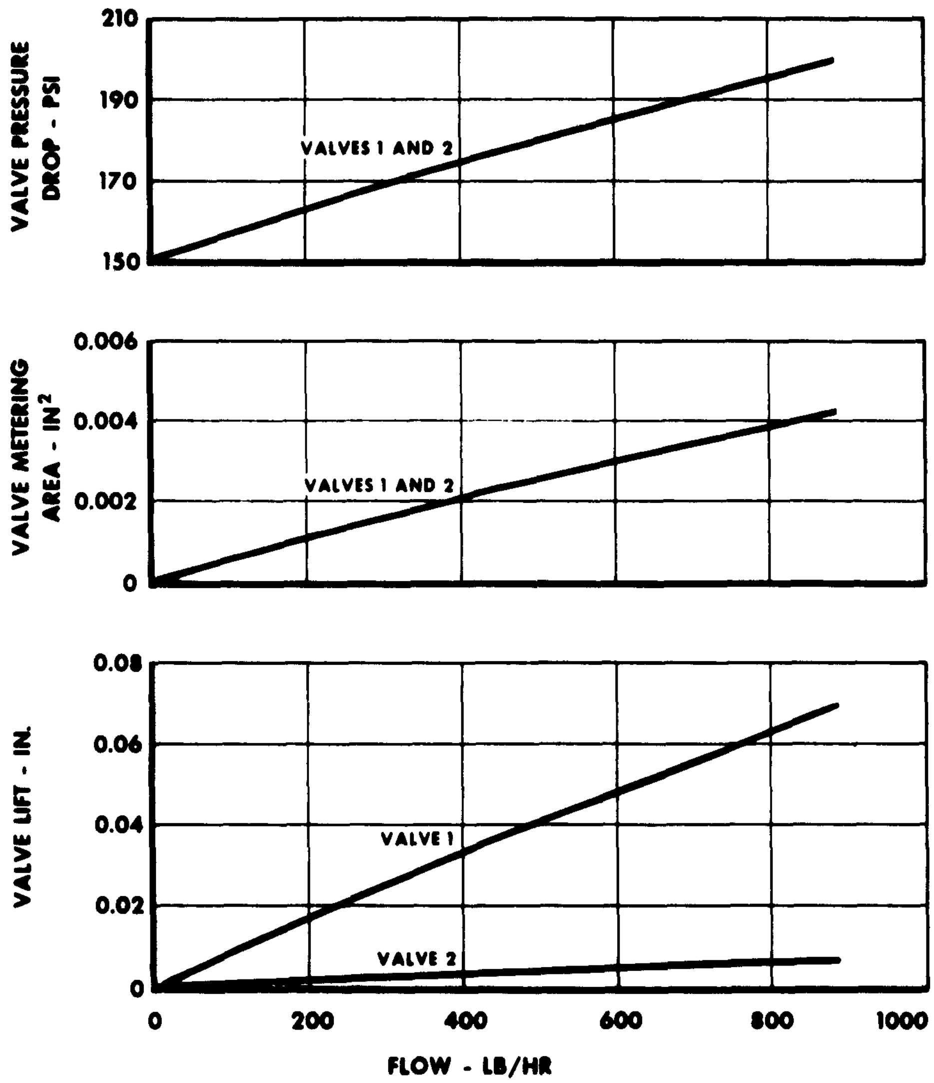


Figure 94 Flow Divider Valve Comparison *FD 3517*

Valve 2 was designed by selecting a slot width of 0.030 inch. This passage size was found to be adequate during the comparative nozzle spray test. This selection determined the required variation of valve lift. This valve characteristic is also shown plotted with pressure and valve opening area in figure 94. Contaminants would be less likely to accumulate in the opening of this valve because of its shape. In addition, if hard materials are used, the cylinder and the slots could be made with sharp corners so that any contaminant that did accumulate would be sheared away.

In valve 2 there would be no positive seat; a very small clearance between the piston and the cylinder could be used to control the flow. To indicate how successful an arrangement using this principle would be, the leakage

flow at 150 psi has been calculated for a valve with a piston diameter of 3/16 inch and several radial clearances. These values are presented below:

Radial Clearance, inch	Leakage flow, lb/hr
0.0002	21.8
0.0001	10.9
0.00001	1.8

It should be recalled that with a clearance of 0.00018 inch in the variable area dual orifice nozzle there was no hysteresis during the comparative nozzle spray tests; consequently so long as the clearance is smaller than this value, the contaminant should not cause poor valve performance.

It should also be noted that the leakage flow of 1.8 lb/hr would not be troublesome for this particular flow schedule because the nozzle would not be used at pressures below 150 psi. For other applications, pressure as high as this would not be required and consequently the leakage flow with a given clearance would be less. As a result the principle is considered to be practical.

d. Spill Nozzle

The spill nozzle can be designed to meet the specifications of 150 psi at 34 lb/hr and the atomization requirement at this low flow only with a high fuel supply pressure or a large inlet fuel flow. This is illustrated in figure 95 where the nozzle inlet flow requirements to achieve a spray with a mean droplet size of 150 microns at a nozzle discharge flow of 34 lb/hr are plotted against the orifice discharge coefficient for two inlet supply pressures, 450 and 750 psig. It can be seen that if a supply pressure of 450 psig is used the fuel flow requirements are of the order of 100,000 lb/hr which is unreasonably high. For an inlet pressure of 750 psig, the required fuel flow is more reasonable. Delavan Manufacturing Company recommended a spill nozzle with an inlet pressure of 750 psig and with a schedule defined by the curve in figure 96. This design corresponds to a discharge coefficient of 0.49 where the orifice diameter is 0.084 inch, and the total slot area is 0.0064 square inch; these values can be obtained from figure 97.

The spill nozzle dimensions are presented below.

Orifice Diameter, inch	Swirl Slot Dimensions, inch	Number of Slots
0.084	0.028 x 0.028	8

To meet the flow schedule shown in figure 96, it is necessary to have a valve similar to a flow divider valve in the spill return line. Although this valve would not be an integral part of the nozzle, its size must be included since it is required for operation with the spill nozzle only. The flow schedules for this valve are defined by the return flow and the spill pressure. This data can be obtained from figure 96. If a discharge coefficient

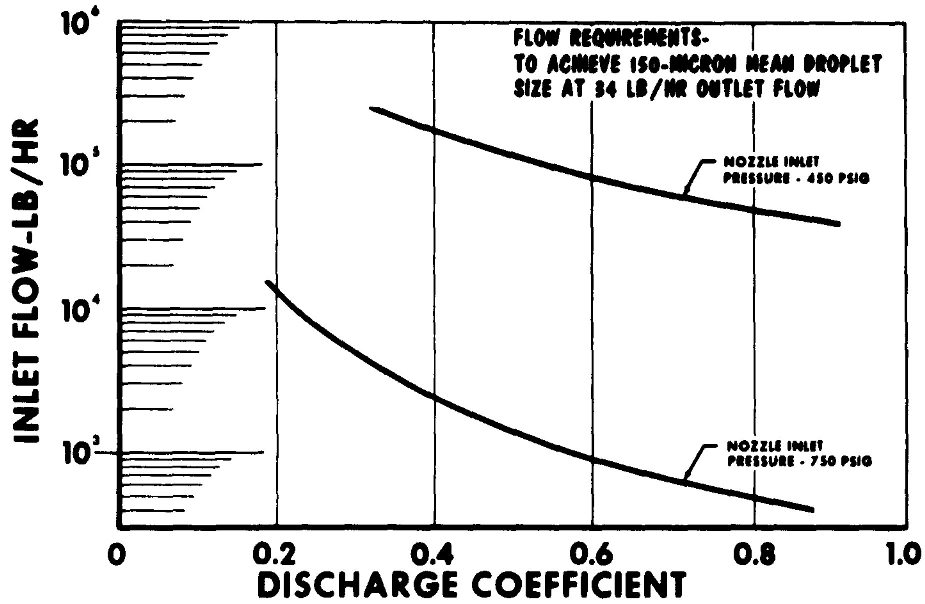


Figure 95 Inlet Flow Requirements, Spill Nozzle *FD 3566*

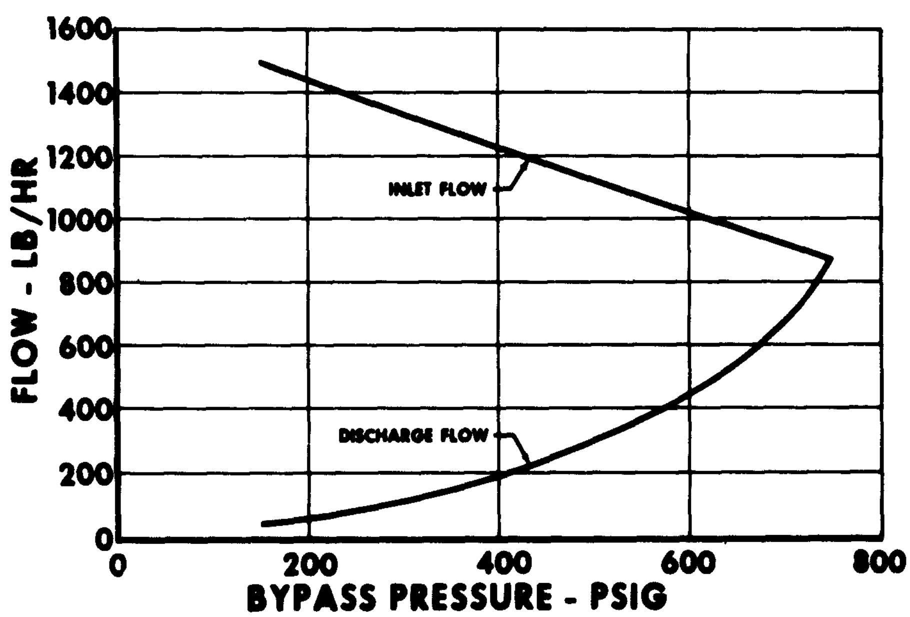


Figure 96 Flow Schedule, Spill Nozzle *FD 3512*

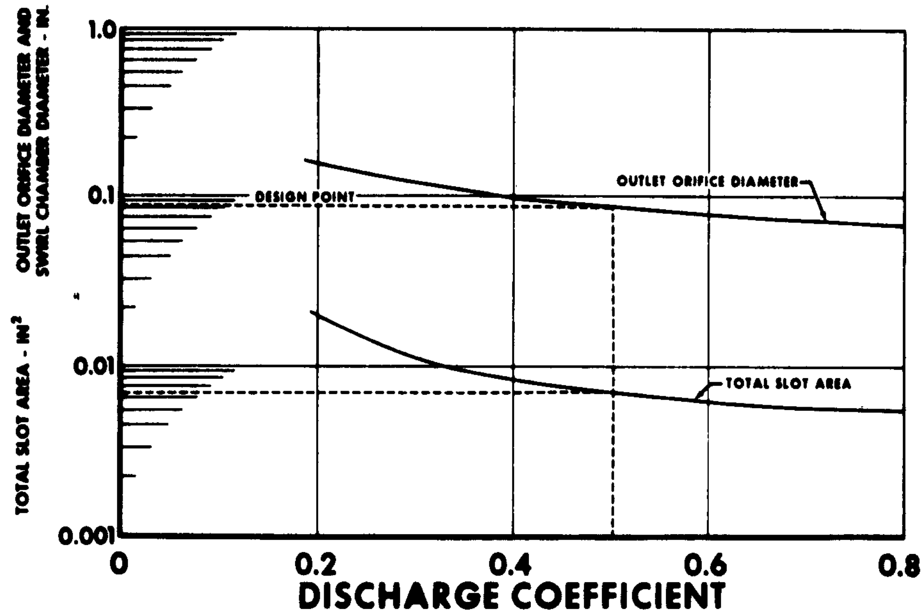


Figure 97 Effect of Discharge Coefficient on Nozzle Dimensions, Spill Nozzle *FD 3565*

of 0.90 is assumed, then the metering area versus nozzle discharge flow can be obtained. Furthermore, if it is assumed that a valve similar to that discussed in the flow divider valve section can be used (a valve in which two slots in the valve piston each with a width of 0.030 inch are used as the metering area), the valve lift is a function of nozzle discharge flow, as shown in figure 98. As shown in this figure, the valve minimum opening size that is defined by the valve lift decreases with nozzle discharge flow.

2. *IMPINGEMENT NOZZLE*

Of the two types of impingement nozzles, the jet-surface impingement type is better for spraying contaminated fuel because its passages would be the largest. For this nozzle type, the nozzle dimensions and the energy requirements were computed. These are discussed below.

Since the impingement nozzle is basically a pressurizing nozzle then it is possible to meet the flow specifications with two different size impingement nozzles and a flow divider valve. If only one impingement nozzle was used, and if the nozzle was designed to meet the minimum pressure requirement of 150 psig at 34 lb/hr, then at the maximum flow of 890 lb/hr the pressure drop across the nozzle would be 10,300 psig. This pressure, of course, is impractical; it indicates the necessity of a two orifice system in which a larger orifice is used for the higher flow rates. The flow to the larger orifice would be controlled by a flow divider valve.

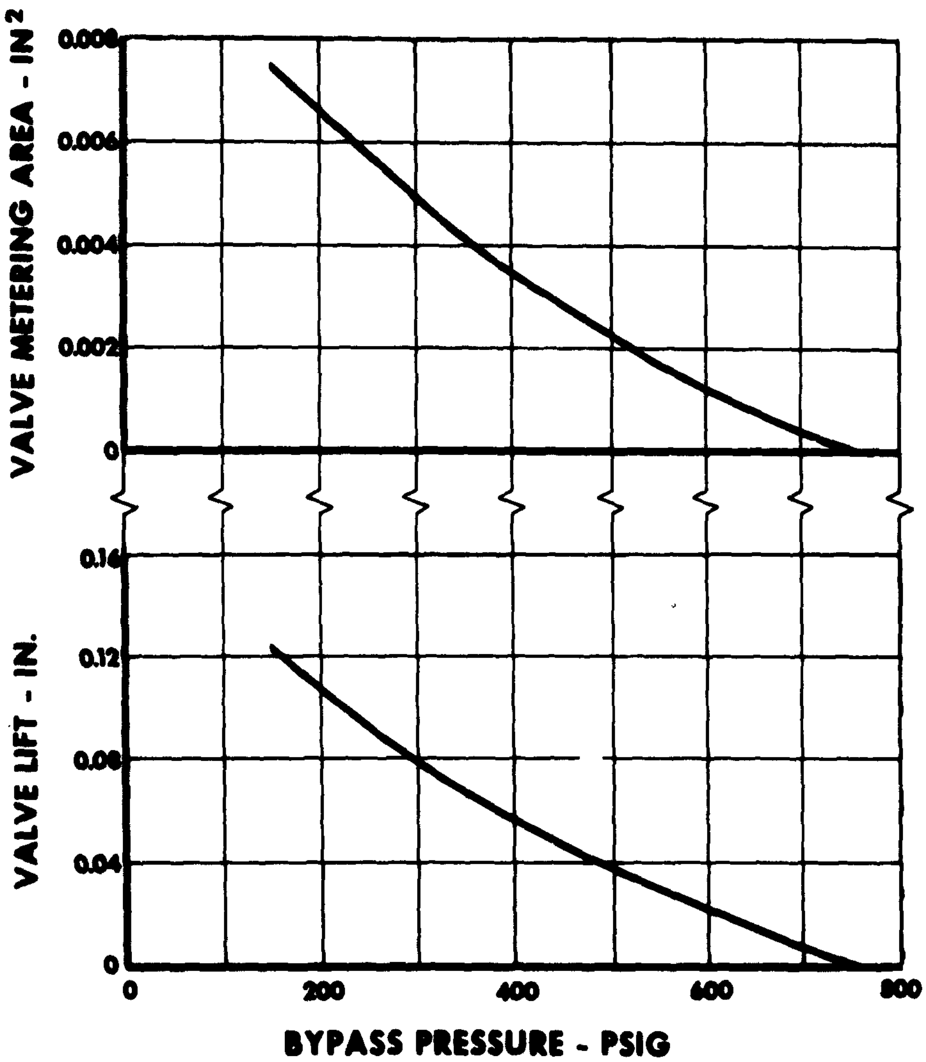


Figure 98 Requirements for Metering Valve in Bypass Line, Spill Nozzle *FD 3508*

An impingement nozzle was designed for the flow schedule specified. The equations presented in Appendix D for the impingement nozzle were used. The primary orifice of this nozzle was sized by assuming a discharge coefficient of 0.90 and by use of the pressure requirement of 150 psi at the minimum flow of 34 lb/hr. The diameter size obtained for these conditions is 0.0154 inch. With an angle of impingement of 45 degrees and the orifice located 1.0 inch from the impingement surface, the droplet size for this size orifice varies as shown in figure 99. (The droplet size varies with impingement angle and the distance from the impingement surface; however, the values cited were selected because these are considered practical.)

In sizing the secondary orifice, the diameter was selected that would produce a spray with a mean droplet size of 150 microns at 240 lb/hr (as in the design of the secondary orifice for the dual orifice nozzle). In figure 100, the design point is illustrated where orifice diameter is plotted against mean droplet size and orifice pressure drop for a flow rate of 240 lb/hr. (The mean droplet diameter that is presented is for an angle of impingement of 45 degrees and the orifice located 1.0 inches from the impingement surface.) The variation of pressure and droplet size with flow rate for this orifice diameter is shown in figure 99.

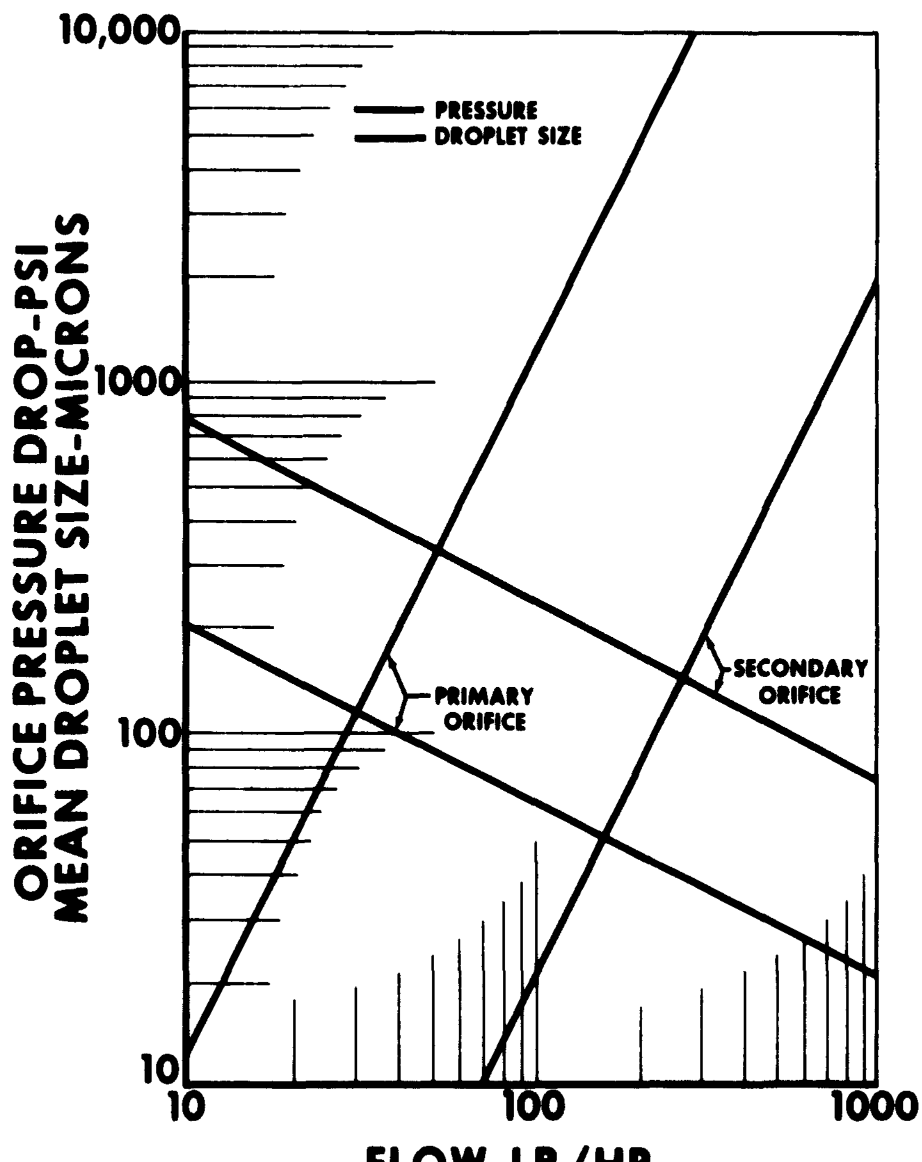


Figure 99 Effect of Flow Rate on Pressure and Mean Droplet Size, Impingement Nozzle FD 3551

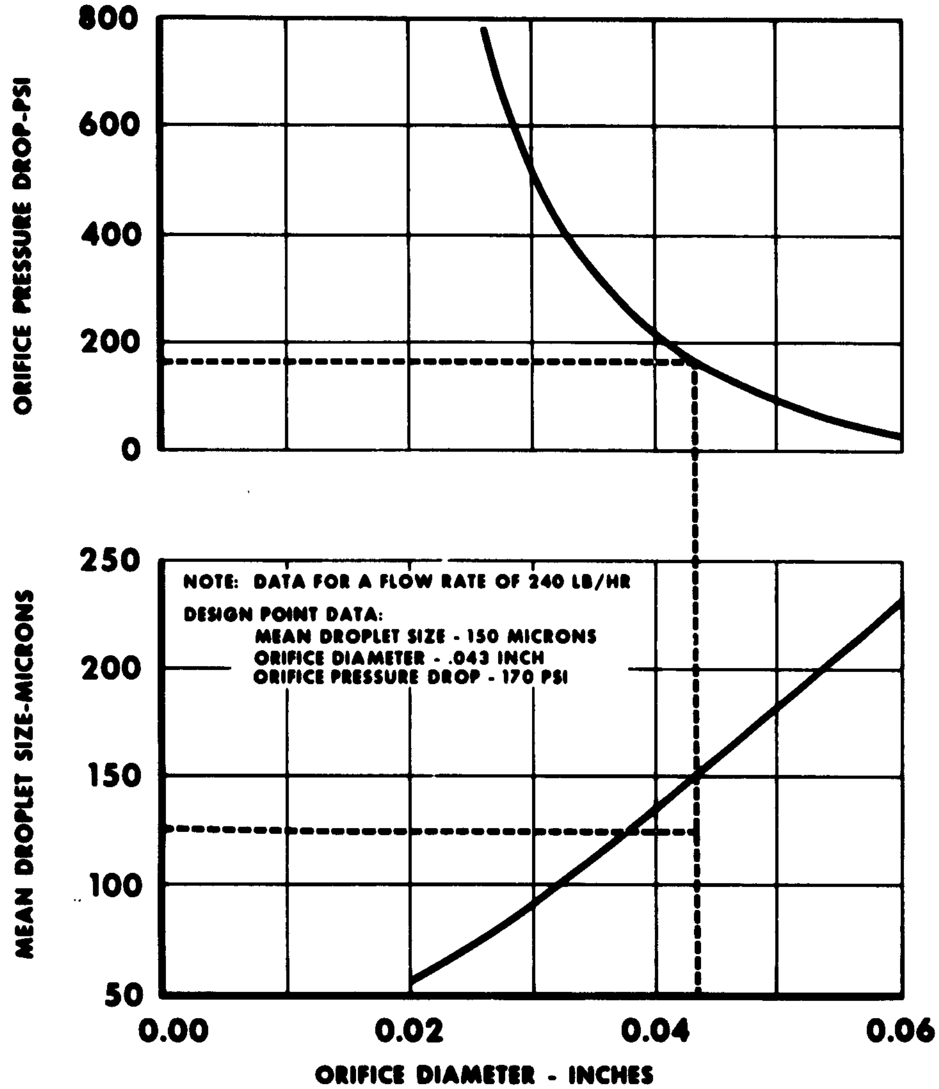


Figure 100 Design Point for Secondary Orifice of Impingement Nozzle *FD 3556*

The flow requirements for a flow divider valve that will schedule the flow through the secondary orifice of the impingement nozzle are the same as those for the dual orifice nozzle. With this specification the flow schedule of the impingement nozzle (orifices and valves together) will be as shown in figure 101.

In summary the impingement nozzle has the dimensions presented in table XV.

TABLE XV. IMPINGEMENT NOZZLE DIMENSION

Orifice	Outlet Orifice Diameter, Inches	Distance from Impingement Surface, Inches	Angle of Impingement, Degrees
Primary	0.0154	1.0	45
Secondary	0.043	1.0	45

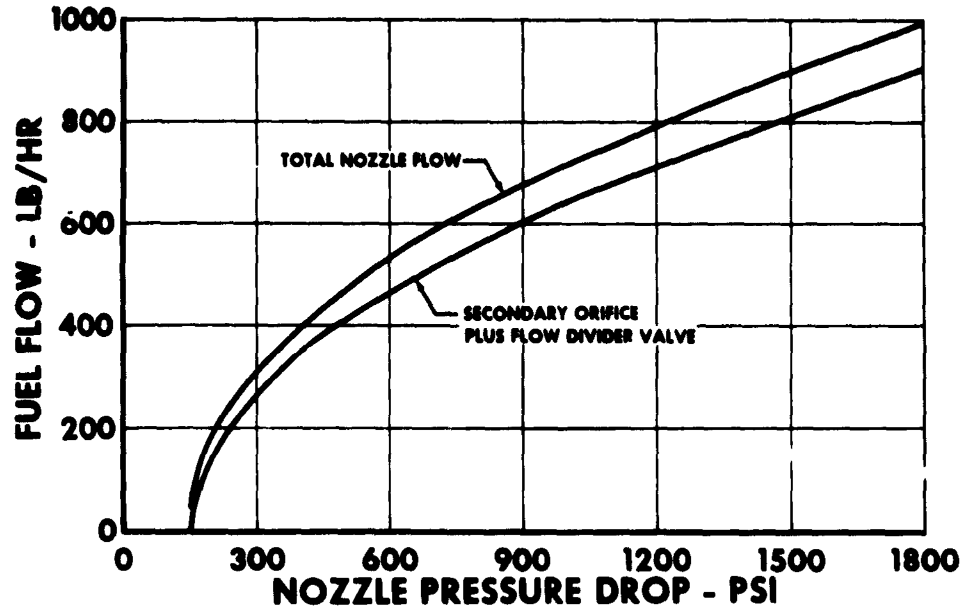


Figure 101 · Flow Schedule, Impingement Nozzle *FD 3504*

3. AIR ATOMIZING NOZZLES

There are three types of air atomizing nozzles that can be used to atomize fuel in a turbojet: (1) internal mixing nozzle, (2) external mixing nozzle, and (3) air assist nozzle. These are described in Appendix C. Of the three nozzles, the air assist is probably the most practical nozzle for turbojet application because (1) moderate pressures are maintained in the fuel system throughout the flow schedule, (2) the spray angle from this nozzle approaches that of a pressure atomizing nozzle, and (3) the airflow requirements are lower than that for other types of nozzles. Because of these characteristics, only an air assist nozzle was considered in the study.

An air assist nozzle for the flow range could be made with a secondary orifice of the dual orifice nozzle enclosed in an air orifice. (The arrangement would be similar to that shown in figure C-11 of Appendix C.) In the air assist nozzle, air would be used to atomize the fuel at low flow rates when the mean droplet size obtained from the nozzle alone is larger than 150 microns. The secondary orifice provides a mean droplet size of 150 microns or less in the flow range 240 to 890 lb/hr. Consequently, air atomization would be required in the flow range from 34 to 240 lb/hr. The air requirements to achieve a mean droplet size of 150 microns have been calculated; these are presented in figure 102.

The dimensions of the air assist nozzle would be the same as those of the secondary orifice of the dual orifice nozzle. However, in the air assist nozzle there could be no primary orifice, and, consequently, no small annulus between the secondary and primary orifices.

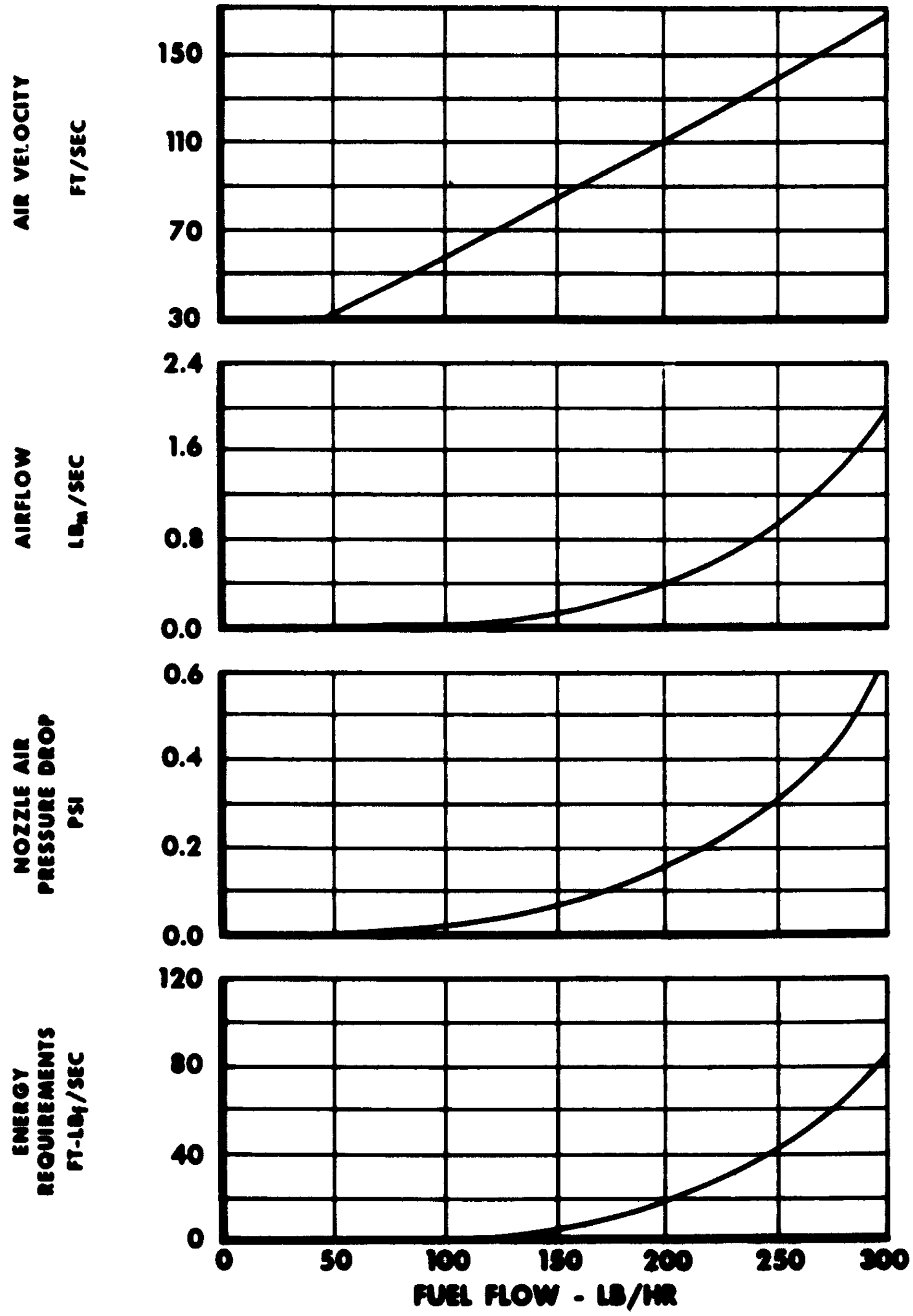


Figure 102 Air Requirements, Air-Assist Nozzle *FD 3395*

4. RESULTS

Shown in figure 103 are plots of the energy requirements of each of the nozzles. The energy requirements for any of the nozzles are not large enough to make their use objectionable. Therefore, the minimum open-

ing dimension will be used as the criterion for comparing the nozzles. The variation of the minimum opening in each nozzle with flow rate is shown in figure 104. The flow divider valve has been considered as a part of the dual orifice nozzle and the impingement nozzle; also the metering valve required in the return line of the spill nozzle was considered to be a part of the spill nozzle. With these valves included in the considerations, a fair evaluation is obtained; if they were not included, some misleading conclusions might be made.

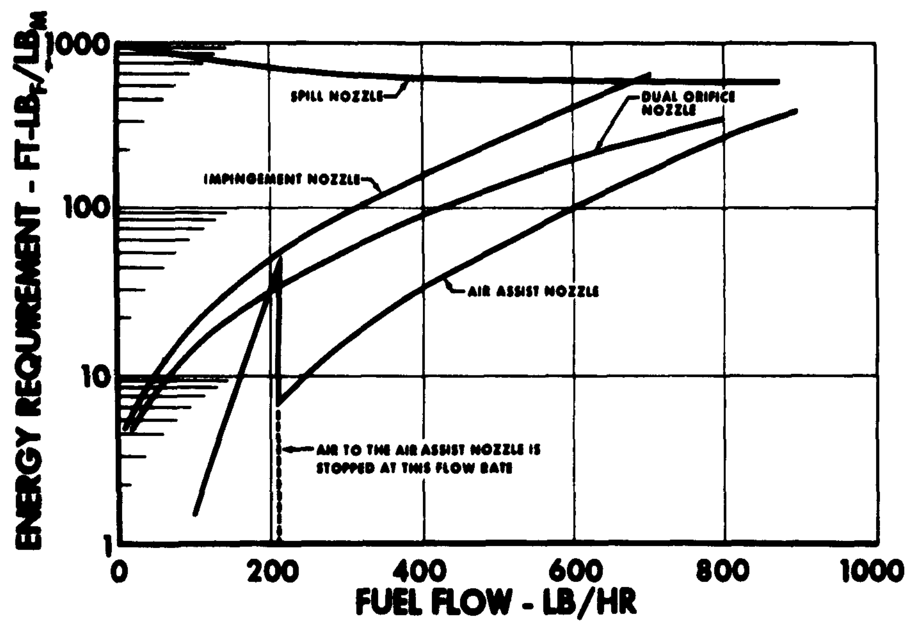


Figure 103 Nozzle Energy Requirements *FD 3570*

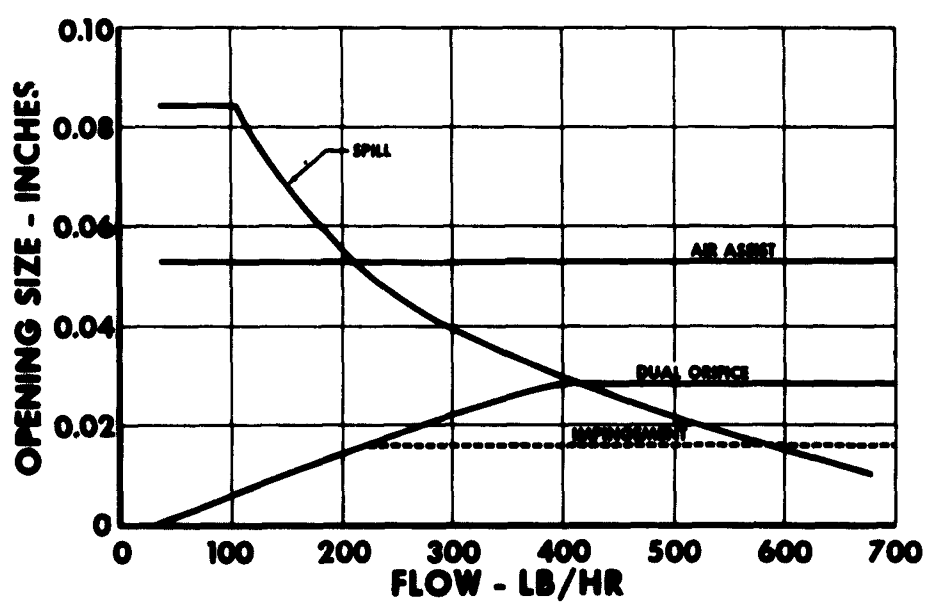


Figure 104 Minimum Nozzle Opening Comparison *FD 3515*

As shown in figure 104, if consideration is given only to passage size, the air assist nozzle is best for spraying contaminated fuel in the flow range considered. However, there are other nozzle characteristics that must be considered. For the air assist nozzle, the characteristics are listed and discussed below:

a. Flow Schedule – The nozzle inlet pressure at minimum flow is much lower than the 150 psi specified. This high pressure requirement at the minimum flow arises from the fact that the fuel is used to cool auxiliary equipment in the aircraft before it is burned. Consequently, the air assist nozzle could not be used without modifying the aircraft so that refrigeration could be obtained from another source.

b. Air Requirements – Although the air requirements are not considered to be excessive, an air assist nozzle has never been used on a turbojet engine. Furthermore, utilization of air assist nozzles could require extensive engine modifications. The design burner can would have to be changed since the air distribution to the can would be changed.

c. Spray Pattern – The spray angle and spray uniformity of an air assist nozzle have not been investigated thoroughly. Furthermore, the repeatability of the spray pattern of the air assist nozzle has never been established.

d. Atomization At Low Fuel Temperature–The energy requirements presented in figure 103 are for a fuel temperature of 80°F. If the temperature is low, the degree of atomization for the same air conditions would be reduced significantly. This would result in poor starting reliability.

Because of these considerations, a feasibility program would be required before the air assist or any type of air atomizing nozzle could be used in a turbojet.

The orifice size of the impingement nozzle alone makes it undesirable. In addition, the spray pattern emitted from this nozzle is completely different from the hollow spray cone of swirl type nozzles that are commonly used in turbojets. The burner can would have to be designed for use with impingement nozzles.

The metering valve in the return line of the spill nozzle decreases with nozzle discharge flow, making the desirability of this nozzle questionable. Another undesirable feature of the spill nozzle is the elaborate control system that is required.

Based on the foregoing considerations, the dual orifice nozzle is the best choice for spraying contaminated fuel in the flow range from 34 to

890 lb/hr, when a minimum pressure of 150 psi must be maintained upstream of the nozzle. In the comparative nozzle spray test the smallest opening in which contaminant did not accumulate is 0.027 inch. Because this size is only slightly smaller than the primary orifice swirl slots it would be desirable to use a 350-micron filter at the nozzle inlet, at least on the primary nozzle.

APPENDIX A

FUNGUS PROPERTIES

This program required that fungus be one of the contaminants in the turbojet engine fuel. Because Pratt & Whitney Aircraft did not have a sufficient quantity of fungus-contaminated fuel, BuWeps supplied the fungus to be dispersed in the fuel. The fungus is mycelium (penicillin mash) from the Pfizer Corporation, Groton, Connecticut. Early in the program, considerable difficulty was experienced in blending the fungus with the fuel. Attempts to blend the fungus in the fuel by stirring, agitation in a Waring blender, and ultrasonic agitation were completely unsuccessful. However, it was found that the fungus blends readily with water, and a good representation of the fungus that will be encountered under actual conditions can be obtained by mixing water with the fungus before dispersion into the fuel. Consequently, it was decided to mix 10 parts by weight of water with 1 part by weight of fungus in the "as-received" condition. The fungus and water mixture was used to replace the water specified in MIL-E-5007B for the contaminated fuel tests. The water in the mixture was the type specified in MIL-E-5007B. This decision was approved by Mr. H. E. Jackson of the Bureau of Naval Weapons during his inspection of the program and facilities at the Florida Research and Development Center on 6 September 1961.

To permit future duplication of tests, an analysis of the fungus in "as-received" condition was made. The following properties were determined:

Density	1.069 grams per cubic centimeter
Water Content	80 percent by weight
Ash Content	1.5 percent by weight

APPENDIX B

CONTACT WITH NOZZLE MANUFACTURERS

1. INTRODUCTION

At the start of this study, several nozzle manufacturers were contacted to obtain information that is pertinent to this program. (See Table B-1.)

The following items were discussed with representatives of each company: (1) work that has been done toward developing a nozzle for spraying contaminated fuel, and (2) nozzles that are currently available that might be evaluated in the nozzle spray tests. Included below are (1) a table showing the nozzle manufacturers that were contacted, (2) a review of the previous work that has been done with contaminated fuel, (3) a list of the reports received, and (4) the nozzles that were recommended for the test.

TABLE B-1. NOZZLE MANUFACTURERS CONTACTED

Firm	Persons Contacted	Date of Visit
Delavan Manufacturing Co.	Rothwell, Tate, Brucker, Burgess, Slezak	18 July 1961
Ex-Cell-O-Corp. Martin Plant	Weldy, Helmrick	19 July 1961
Parker-Hannifin Corp. Accessories Division	Webster, Simmons, Cleminshaw	20 July 1961
Eddington Specialty	Czarnecki	21 July 1961

In addition, the program was discussed with Mr. J. F. Campbell of the Campbell Development Corporation, Gates Mills, Ohio during his visit to FRDC on 23 December 1961. Mr. Campbell designed the variable area dual orifice nozzle that was evaluated in the comparative nozzle spray test.

2. REVIEW OF PREVIOUS WORK WITH CONTAMINATED FUEL

It was found that Ex-Cell-O, Delavan, and Parker have performed tests with contaminated fuel. However, fungus and algae were not included as constituents of the contaminant in any phase of these experiments. The work at Ex-Cell-O was done to develop a nozzle that would resist erosive effects of the contaminants. Delavan and Parker have evaluated nozzles designed for spraying fuel contaminated in excess of that specified by MIL-E-5007B, in addition to erosion work. Both of these companies stated that the cotton lint is the most troublesome contaminant constituent because it accumulates in the corners of the mesh of a filter and in the nozzle passages, thus obstructing flow.

In addition to the general review presented above, Delavan Manufacturing, who was a subcontractor for a portion of the program, included a detail review of all of their work with contaminated fuel in their first progress report (Delavan Technical Report No. 196). This review is presented below:

a. The modulating characteristics of a dual orifice nozzle with a flow divider valve was evaluated while spraying fuel contaminated with MIL-E-5007A contaminant. The following conclusions were made:

   (1) Contaminant particles in the fuel cause erratic modulation of nozzles with integral flow divider valves when operating in the transition range.

   (2) The detrimental effects of contaminated fuel upon modulation in the transition range can be greatly reduced by the combination of (a) a 10-micron filter upstream from the nozzle and (b) the application of vibration to the nozzle.

   (3) There is evidence that operational time on contaminated fuel can be extended by periodically clearing the nozzle with abrupt surges of high operating pressures.

b. In another study, an evaluation was made of the performance of three dual orifice nozzles with integral flow dividers using contaminated fuel. That portion of the flow range in which transition from primary to combined primary and secondary flow occurs was studied in particular. From these tests Delavan concluded that MIL-E-5007A contaminant would cause malfunction only in a flow divider type of nozzle, and then only at the secondary cutin.

c. A study was made of afterburner fuel injectors in which fuel was subjected to "Normal" (MIL-E-5007B through a 74-micron filter) and "Emergency" (MIL-E-5007B) contamination tests. The conclusions of that study were:

   (1) The strainer opening sizes were large enough to prevent plugging and to protect the metering mechanism against the unusually large objects or metal chips.

   (2) The injectors evidenced some plugging during the test; however, they were able to clean themselves and continue flowing.

d. A 50-hour endurance test was conducted on flow divider valves with MIL-E-5007B contaminants that included lint. The lint proved to be a definite plugging factor.

e. In another study three injectors were evaluated to compare contamination (MIL-E-5007B) buildup in the nozzle when using different strainers. A 74-micron filter and a 150-micron filter were evaluated. It was found that an increase of strainer size from 74 microns to 150 microns resulted in nozzles passing a 20-hour endurance test, whereas the 74-micron strainer plugged after 5 hours.

f. Another test was run to determine whether a strainer with larger openings would satisfactorily pass the MIL-E-5007A test. A 50-mesh screen with a 0.0085-inch wire diameter (300 microns) was used in the test. The results showed that the strainer retained lint that in turn was capable of restricting particles smaller than the strainer openings.

g. Six Delavan fuel nozzles were tested with fuel contaminated with MIL-E-5007B filtered through a 74-micron filter for 10 hours. After the test the nozzles were flow tested, disassembled, and evaluated. The flow schedules of the nozzles were not changed significantly by the test.

3. REPORTS RECEIVED

a. Contaminated Fuel Reports

The following reports, which summarize work with contaminated fuels, were received from Delavan Manufacturing Company:

(1) O'Hara, R. J. "Evaluation of DLN 6525A Fuel Nozzle Assembly After Contamination Test," Delavan Manufacturing Co. Technical Report No. 128, July 16, 1959.

(2) Wilcox, R. L. "Fuel Contamination Study, DLN 8275 Fuel Nozzle Program," Delavan Manufacturing Co. Technical Report No. 145, December 29, 1959.

(3) Brucker, J. R. "Evaluation of DLN 8275 Nozzles After Contaminated Fuel Endurance Tests," Delavan Manufacturing Co. Technical Report No. 146, December 30, 1959.

A summary of these reports is included in Section 2, preceding.

b. Fuel Nozzle Reports

In addition to the foregoing reports, some reports were received that were helpful in the analytical studies of this contract. These reports are listed below under the supplier.

(1) Delavan Manufacturing Company

(a) "Atomizing Nozzles for Combustion Applications," published by Delavan Engineering Department, February 1961.

(b) "Spray Droplet Technology," published by Delavan Engineering Department, March 1958, revised January 1961.

(c) Tate, R. W., "Spray Patternation," reprinted from Industrial and Engineering Chemistry, Vol 52, page 49a, October 1960.

(2) Accessories Division of Parker-Hannifin Corporation

(a) Webster, W. G. "A Guide for the Selection of Fuel Nozzles for Gas Turbine and Similar Applications," (no date)

(b) Simmons, H. C. "Design Report on High-Flow-Capacity Small-Diameter Variable-Area-Nozzle," Engineering Report No. 1341-R5270, November 29, 1960.

(c) Simmons, H. C. "The Spill Nozzle as Used in Gas Turbine Fuel Injection System," Engineering Report No. 1343-Q-5248, May 24, 1960.

4. NOZZLES RECOMMENDED FOR THE COMPARATIVE NOZZLE SPRAY TESTS

The following nozzles were recommended for the comparative nozzle spray tests by the respective nozzle manufacturers. (The nozzles that were selected for the test are described in Section V of the report.)

a. Delavan Manufacturing Company

(1) Dual orifice nozzle which is presently used on the J75 engine

(2) Variable area nozzle

(3) Simplex nozzle

(4) Spill nozzle

(5) Industrial air atomizing nozzle.

b. Accessories Division of Parker-Hannifin Corporation

(1) Dual orifice nozzle with integral flow divider made of special materials to resist corrosion by contaminants and having features to minimize clogging

(2) Dual orifice nozzle with integral flow divider with an alternate approach of that used in Nozzle No. 1 for minimizing corrosion

(3) Parker-pintle nozzle

(4) Spill nozzle

c. Eddington Metal Specialty Company

(1) Variable area nozzle-Eddington Designation No. V-181.

d. Ex-Cell-O Corporation-Martin Plant

(1) Dual orifice with external flow divider which is presently used on the J57 engine

(2) Dual orifice with external flow divider which is presently used on the JT12 engine.

APPENDIX C

DESCRIPTION OF NOZZLE TYPES

The three basic atomizing nozzles, which were (1) the pressure atomizing, (2) the air atomizing, and (3) the impingement atomizing, are discussed below. This information was taken from the Delavan Manufacturing Co. Progress Reports, Delavan Technical Reports 196 and 199.

1. PRESSURE ATOMIZING NOZZLES

a. *THE SIMPLEX NOZZLE*

The simplex nozzle is the simplest, pressure atomizing, swirl type nozzle. Figure C-1 shows the basic design of this nozzle. The liquid is forced through the tangential distributor slots under pressure and enters the swirl chamber at a high velocity. The liquid then moves toward the orifice at the center of the swirl chamber in a spiral path as a free vortex. (The velocity of the liquid increases as it approaches the center of the vortex.) The liquid emerges from the orifice in a conical film of liquid that breaks up into droplets of varying size. For a given simplex nozzle, variation in discharge rate is proportional to the square root of the nozzle pressure.

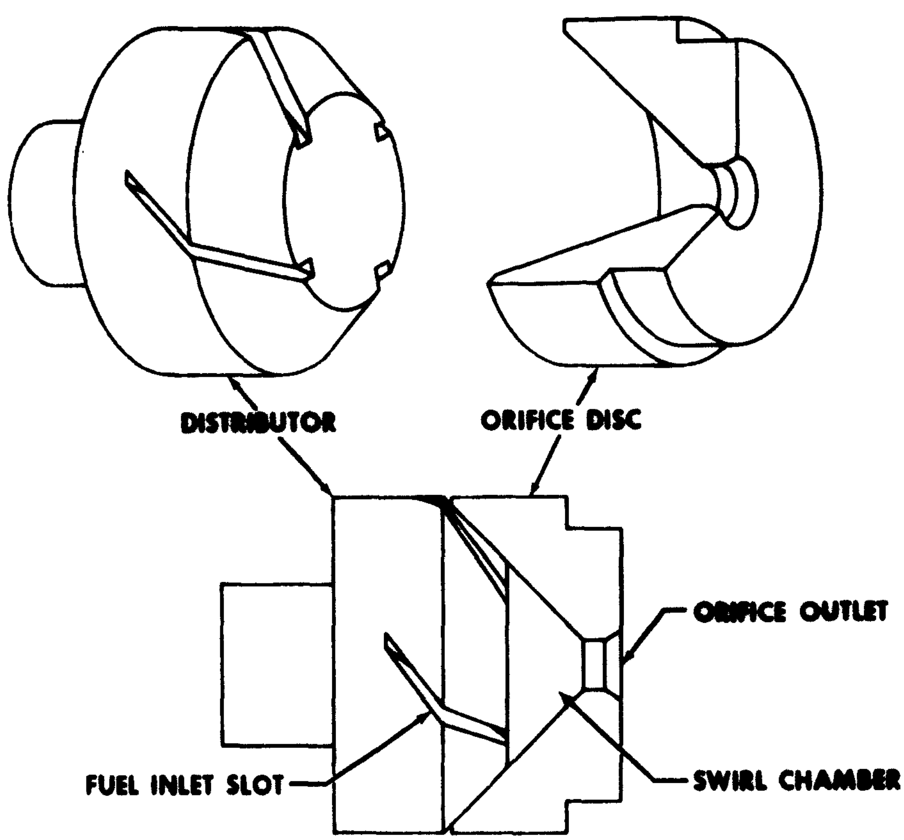


Figure C-1 Simplex Nozzle Diagram *FD 3417*

The droplet size will be different for nozzles of different flow rates and spray angles. The curve in figure C-2 shows the variation in mean droplet diameter for nozzles of different sizes for a nozzle pressure drop of 100 psi. Droplet size is generally larger for higher capacity nozzles, all other conditions remaining constant.

Simplex nozzles and other swirl chamber nozzles generally produce a hollow spray cone. Spray angle is a function of the geometry of the metering passages such as orifice length, orifice contour, swirl chamber design, slot design, and ratio of slot area to orifice area. It is extremely difficult to construct a simplex nozzle with a spray angle (included angle measured at the orifice) less than 30 degrees. At high flow rates narrow angles are difficult to produce.

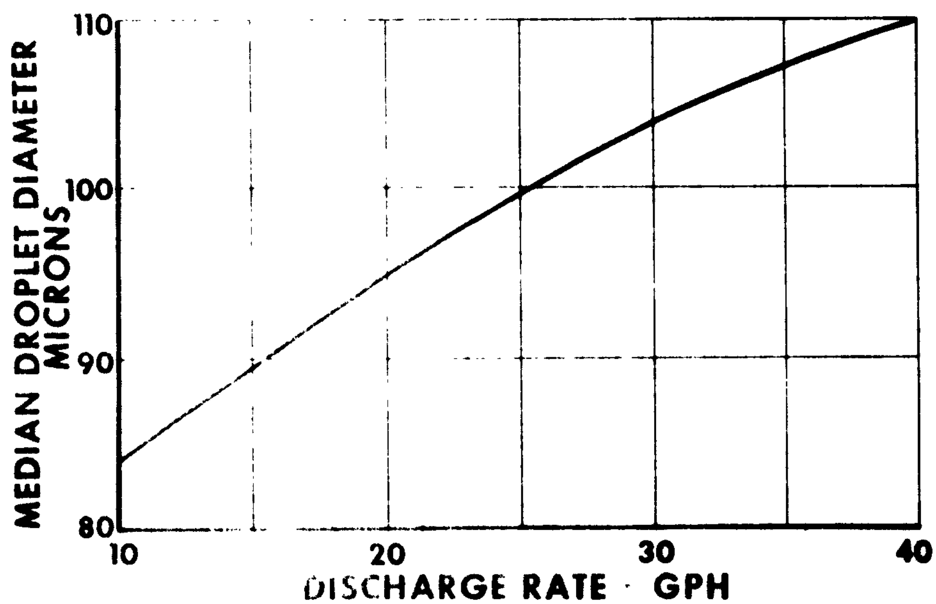


Figure C-2 Droplet Diameter vs Flow Rate **FD 3554**

b. THE DUAL ORIFICE NOZZLE

A dual orifice nozzle consists of a nozzle within another nozzle. Essentially, a dual orifice nozzle is two concentric simplex nozzles within one body, (1) an inner, low capacity nozzle called the primary stage, and (2) an outer, annular nozzle called the secondary stage. This design provides good atomization, a wide flow range, and a uniform spray angle over the entire range (characteristics particularly desirable in gas turbine applications).

A schematic of a dual orifice is shown in figure C-3. At low flows, the dual orifice nozzle operates as a simplex nozzle with only the primary orifice flowing. When the flow from the primary orifice is stable and well atomized, fuel is admitted to the secondary orifice. The introduction of the secondary flow is usually delayed until after the fluid pressure of the primary system has increased to the extent that the primary spray can assist with the breakup of the secondary spray during the phase when secondary pressure is less than 10 psig.

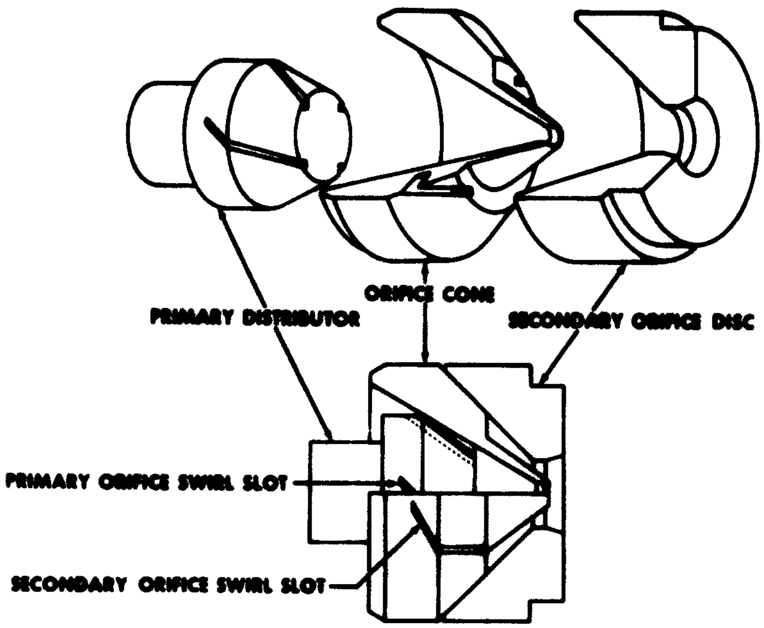


Figure C-3 Dual Orifice Nozzle Diagram *FD 3560*

A flow ratio of 20 to 1 is quite common with dual orifice nozzles, and nozzles with a 50 to 1 ratio (for example, a flow range of 20 to 1000 lb/hr) have been built. The minimum primary flow determines the dual orifice nozzle minimum flow requirement. Figure C-4 illustrates a typical flow curve for a dual orifice nozzle.

A valve is required to "schedule" flow to the secondary orifice. Both separate external flow divider valves and integral flow divider valves are used.

Specific information on spray droplet size from dual orifice nozzles will vary with the design configuration. The droplet size from the primary nozzle will be equivalent to that from a simplex nozzle of the same capacity and at the same pressure. As secondary flow is introduced and energy from the primary spray is used to atomize the secondary orifice flow, the droplet size

will be larger. At maximum discharge, the spray characteristics are approximately the same as the simplex nozzle with the same discharge rate and the same pressure.

A relatively constant spray angle can be maintained over the entire operating range of dual orifice nozzles. Spray angles within the 70° to 90° range are possible.

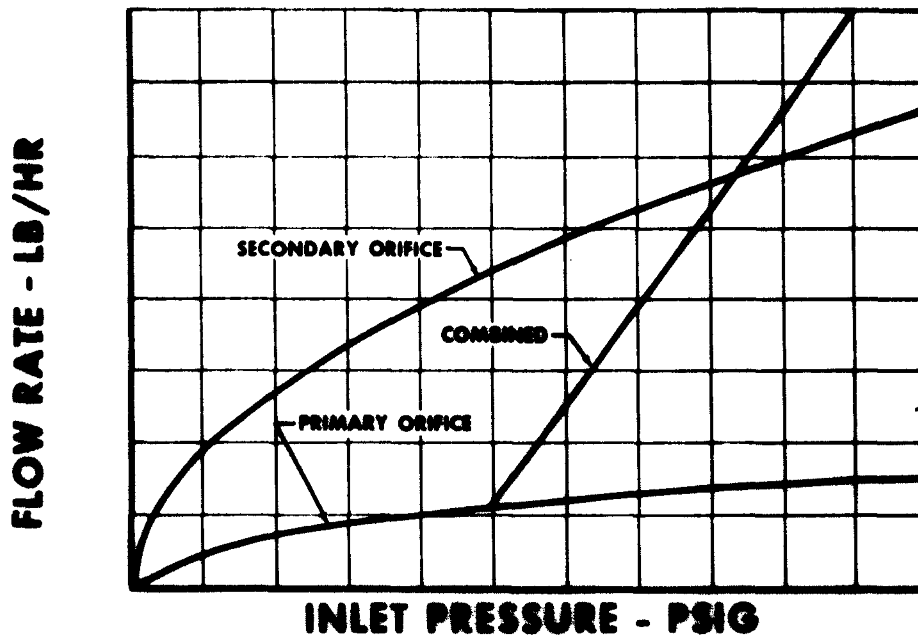


Figure C-4 Typical Dual Orifice Nozzle Curve *FD 3533*

c. *THE DUPLEX NOZZLE*

A duplex nozzle is a nozzle with a single orifice and two sets of inlet swirl slots. A duplex nozzle is shown schematically in figure C-5. The flow through both sets of swirl slots is combined before the liquid is discharged through the orifice. The flow to the large set of swirl slots is varied by a flow divider valve similar to that used with the dual orifice nozzle.

The nozzle orifice is sized for the maximum discharge rate. Because of its large orifice size, the duplex nozzle does not provide as small a droplet size or as stable a spray pattern at low discharge rates as the dual orifice nozzle. This, together with the extremely small dimensions of the primary slots, limits the minimum discharge rate. Assuming equivalent atomization, the duplex nozzle flow range will be considerably less than the dual orifice nozzle flow range.

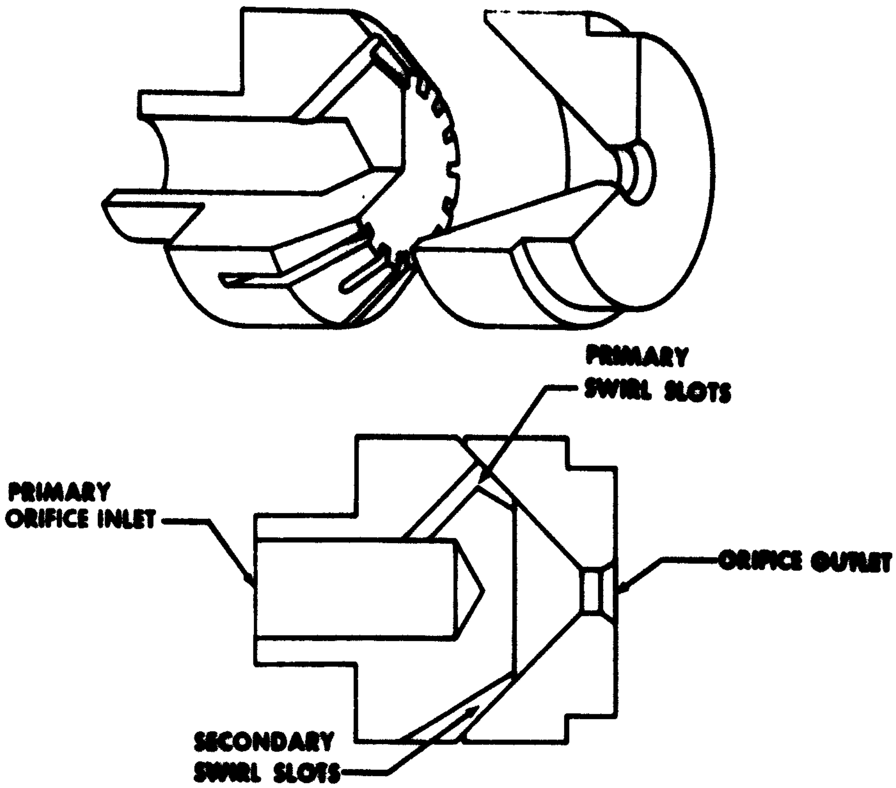


Figure C-5 Duplex Nozzle Diagram *FD 3448*

With a duplex nozzle, the spray angle increases at low fuel flows. Spray angles at maximum discharge rates can be between 45 and 90 degrees.

d. *THE BYPASS NOZZLE*

A bypass nozzle or spill nozzle is shown schematically in figure C-6. The mechanism of atomization is essentially the same as in a simplex nozzle. A wide flow range is accomplished by providing a return flow from the swirl chamber to the suction side of the supply pump. The outlet orifice and tangential slots are the same size as those of a simplex nozzle that would handle the maximum discharge rate. The bypass nozzle maximum/minimum flow ratio is virtually unlimited. Minimum discharge rates with good atomization at 1.5 pounds per hour are common.

A range of spray angles from 30 to 90 degrees is possible at the maximum discharge rate with a bypass nozzle. The spray angle increases as the flow decreases. It is possible to design the nozzle so that this angle change is not excessive. With a flow ratio of 20 to 1, the spray angle at minimum flow may be expected to be 10 to 20 degrees larger than at the maximum discharge rate.

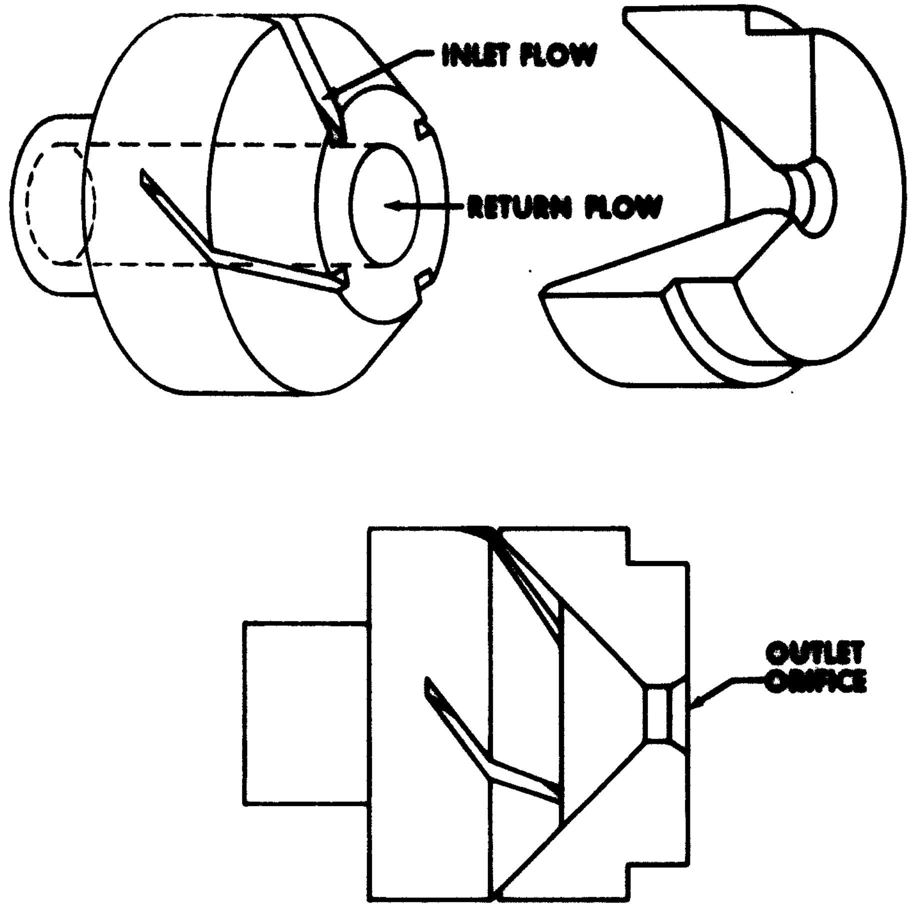


Figure C-6 Bypass Nozzle Diagram *FD 3559*

e. VARIABLE AREA POPPET NOZZLE

This nozzle is distinctly different from the swirl chamber nozzle. A schematic of the nozzle is shown in figure C-7. Atomization is accomplished by spreading the liquid with a poppet into a thin conical film that breaks up into droplets as the film diverges. As the pressure is increased, the poppet moves forward, exposing a larger orifice area and permitting a greater flow through the nozzle.

Satisfactory atomization has been achieved with this nozzle over a range with a flow ratio in excess of 50 to 1. The curve in figure C-8 illustrates a ratio of 28 to 1 with 400 psig maximum nozzle pressure. At present, the minimum flow limitation appears to be 30 lb/hr for a fuel with a viscosity of 1 centistoke.

This nozzle permits spray angles from 30 to 180 degrees. A practical range is 45 to 120 degrees. The spray angle is fairly uniform over the entire range of operation.

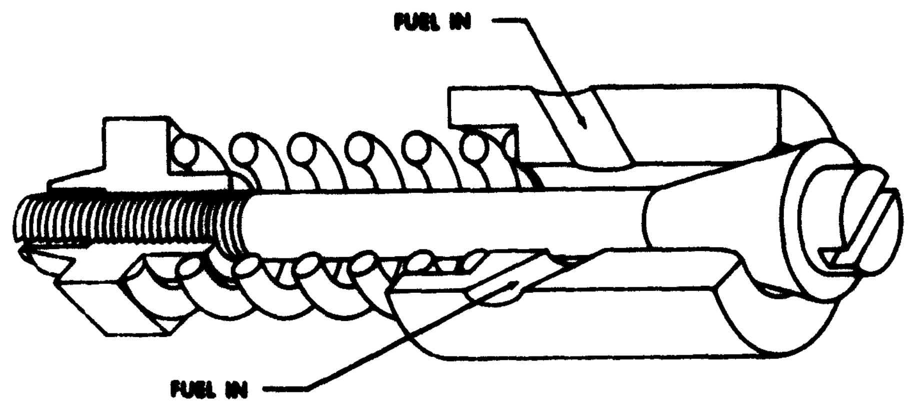


Figure C-7 Variable Area Nozzle *FD 3544*

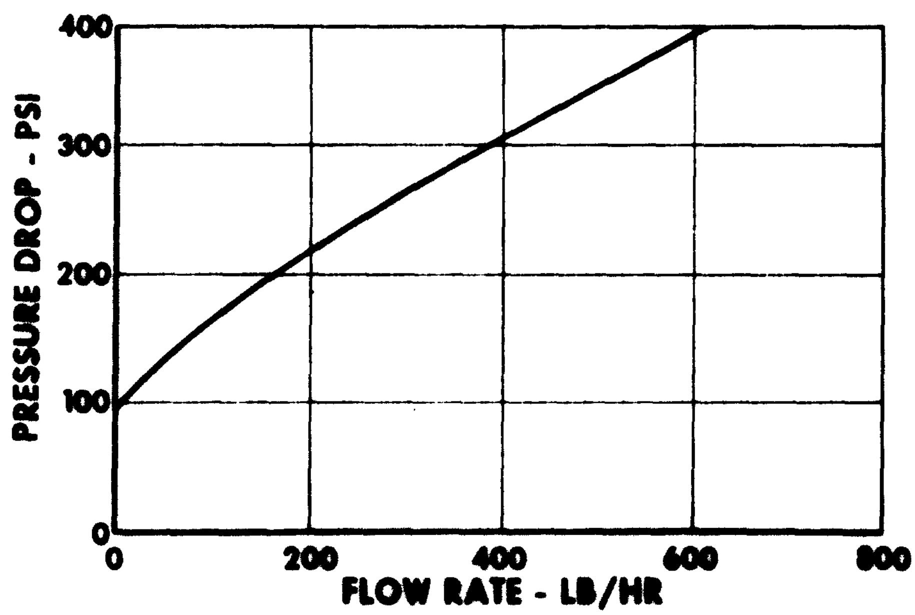


Figure C-8 Typical Flow vs Pressure Curve *FD 3553*

2. IMPINGEMENT NOZZLES

The term impingement nozzle refers both to impact and impinging-jet atomizers. In the former, breakup is caused by a liquid jet colliding against an external surface. In the latter type, two or more jets impinge on each other, resulting in atomization. Neither of these type nozzles have ever been used in turbojets.

a. IMPACT NOZZLES

Impact nozzles have the advantage of no internal parts that can wear or clog. Such a nozzle with a single orifice should be capable of passing contaminants more readily than the multiple orifices which are required for impinging jets. A stable, uniform spray pattern is difficult to achieve with an impact device because the plate or pin interferes with the spray.

b. IMPINGING-JET NOZZLES

Impinging-jet nozzles are used extensively in liquid propellant rocket engines. The jets of this nozzle must be directed very accurately so that they will collide. This requires accurate machining and alignment of the orifices. Another problem is poor atomization at low jet velocities. When the velocities are too low, very large droplets are formed around the periphery of a flat, oval-shaped sheet of liquid. As velocity increases, the sheet disappears and smaller droplets emanate from the point of impingement. (Jet velocities of at least 30 to 40 ft/sec are required for good atomization.)

3. AIR ATOMIZING NOZZLES

In an air atomizing nozzle, the energy for atomization is provided by compressed air instead of fuel pressure. There are three types of air atomizing nozzles: internal mixing, external mixing, and air assist. These are discussed below.

a. INTERNAL MIXING NOZZLE

In an internal mixing nozzle (figure C-9), air is forced through tangential slots and enters the swirl chamber in much the same manner as the liquid in a pressure-atomizing nozzle. The fuel may be supplied either at the tangential slots or upstream from them so that the turbulence in the swirl chamber causes thorough mixing. As the fuel-air mixture emerges from the orifice, the expanding air shears the liquid into very small droplets. The internal-mixing nozzle is not generally a metering nozzle. A common practice is to meter the fuel by an external means that provides a positive fuel flow to the nozzle. The pressure of the fuel at the nozzle is established by the air pressure and fuel flow rate, and it must be balanced so that fuel will not back up into air supply and vice versa.

Spray angles from an air-atomizing nozzle are difficult to define. The spray is essentially a solid cone. A wide spray angle can be produced with a high fuel-air ratio. The minimum spray angle obtainable is approximately 30 degrees, and the maximum spray angle without drooling is approximately 90 degrees.

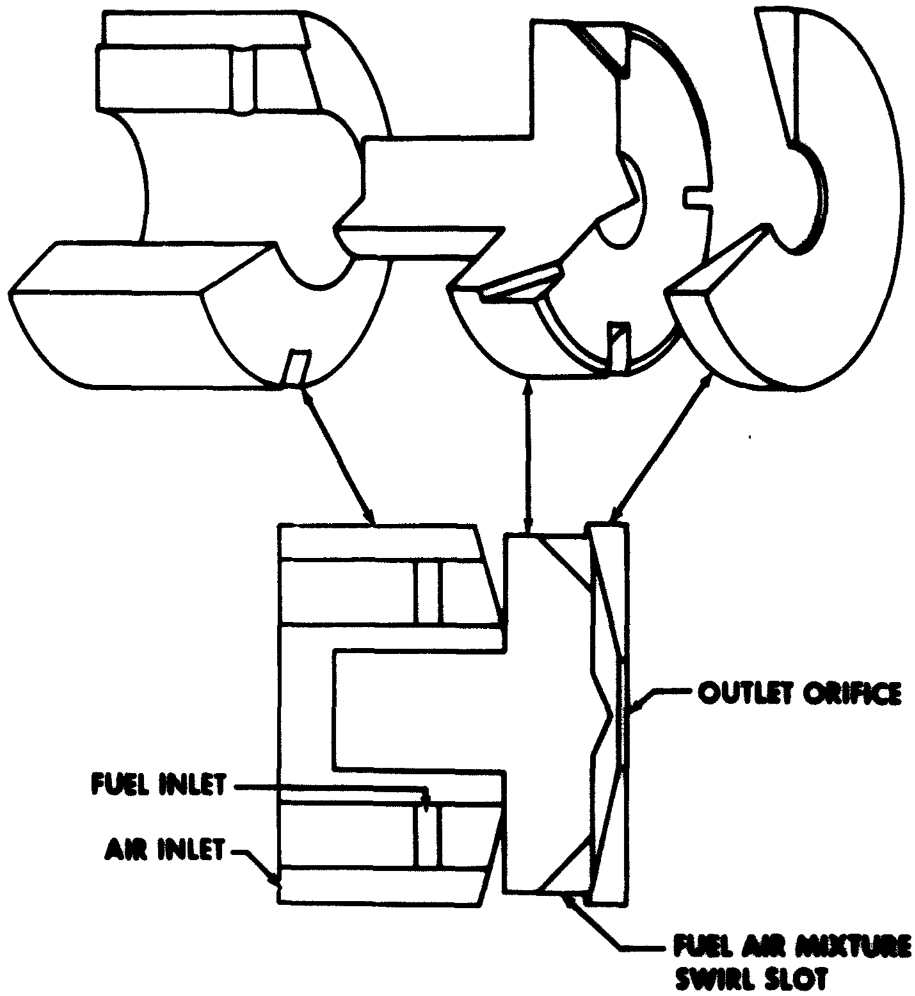


Figure C-9 Internal Mixing Air Atomizing Nozzle *FD 3555*

b. *EXTERNAL MIXING NOZZLE*

In the external mixing nozzle, which is shown in figure C-10, fuel and air passages are not interconnected inside the nozzle. The fuel emerges from the nozzle at low velocity and the air impinges upon the stream of fuel, breaking it into small droplets. This nozzle can be designed to function on a wide range of air pressures. However, in most instances, for satisfactory atomization, a lower fuel-air ratio must be maintained with the external mixing nozzle than with the internal mixing nozzle. With some designs, gravimetric fuel-air ratios of 1 to 1 are used. This nozzle can be used to atomize a wide flow range. Spray angles range from 20 to over 100 degrees, depending upon the nozzle design.

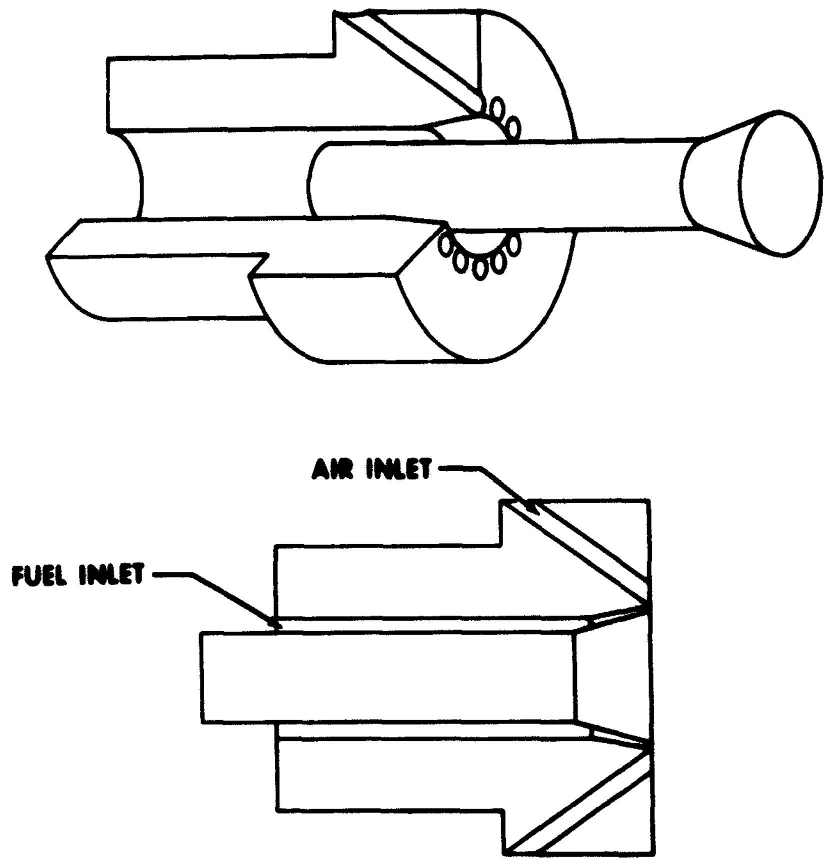


Figure C-10 External Mixing Air Atomizing Nozzle *FD 3564*

c. *AIR-ASSIST NOZZLE*

An air-assist nozzle uses air for atomization at the low fuel flows and operates as a pressure-atomizing nozzle at high fuel flows. This scheme permits good atomization with much lower discharge rates than are possible with a simplex (pressure atomizing) nozzle. By using air only for the lower flows, the total air required for atomization is minimal. A schematic of this nozzle is shown in figure C-11.

With an air-assist nozzle a very high fuel-to-air ratio is possible. This nozzle is a metering device and follows the same flow-pressure relationship as a simplex nozzle. At low fuel flows and, therefore, at low pressures, distribution to several nozzles may have to be controlled by an auxiliary modulating device.

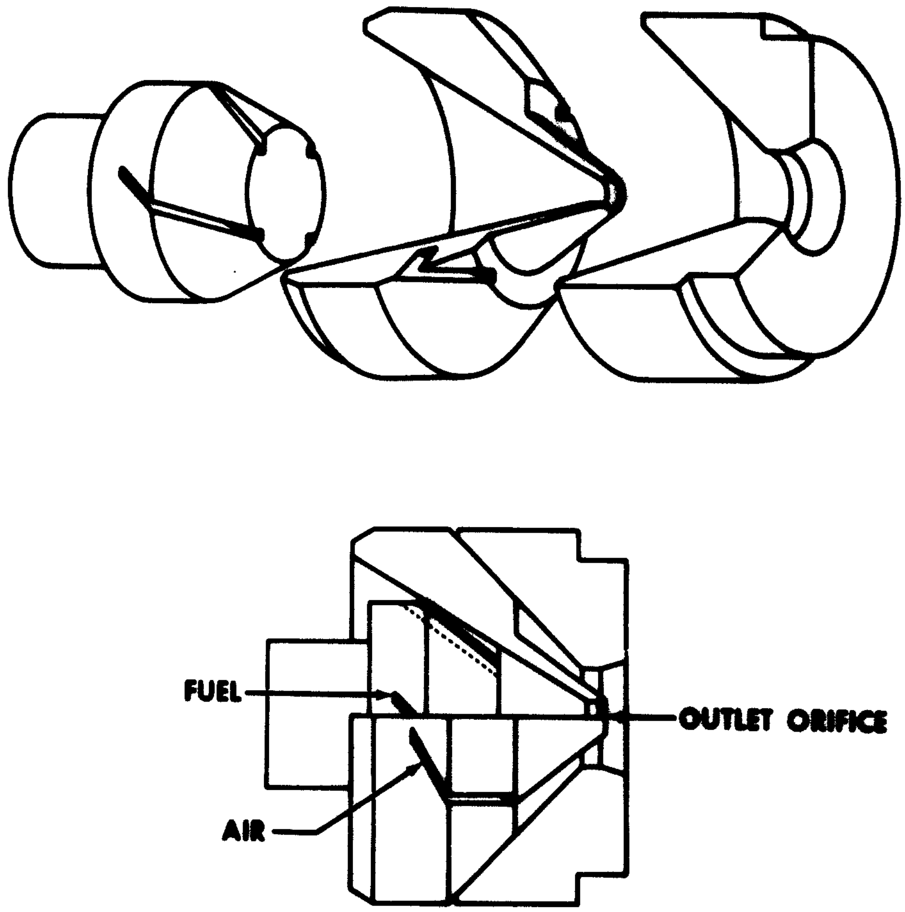


Figure C-11 Air-Assist Nozzle *FD 3532*

APPENDIX D

EQUATIONS USED IN ANALYTICAL NOZZLE DESIGN STUDIES

The equations used during the analytical studies are presented in this section of the appendix. The section is divided according to nozzle type.

1. PRESSURE ATOMIZING NOZZLES

a. THEORY

The dual orifice nozzle and the spill nozzle were designed by use of the curves that were developed by R. W. Tate and W. R. Marshall (Ref. 1). These design curves were derived from the results of tests with water during which the effects of basic nozzle properties on spray characteristics were determined. The three fundamental nozzle variables studied were (1) diameter of the nozzle orifice, (2) the liquid axial velocity component, and (3) the tangential velocity component of the liquid.

The results of the study are empirical expressions for the spray mean droplet size, the spray angle, and the standard deviation of the droplet size, which is a measure of the uniformity of the spray. The results were presented in the form:

$$D = f_D (V_v)\, g_D (V_t)\, h_D (d) \tag{1}$$

$$\theta = f_\theta (V_v)\, g_\theta (V_t)\, h_\theta (d) \tag{2}$$

$$S = f_S (V_v)\, g_S (V_t)\, h_S (d) \tag{3}$$

where

- D = mean droplet size in microns
- θ = plane angle, including 80 percent of the total spray volume in degrees. (Because this value is very nearly the same as the spray angle measured at the outlet of the orifice, it was used during the study.)
- S = standard deviation
- V_v = $0.407Q/d^2$ = vertical velocity component, ft/sec
- V_t = $0.32Q \cos\alpha / N A$ = inlet tangential velocity component, ft/sec
- d = Orifice diameter, in.
- Q = Orifice flow rate, gal./min
- α = angle made by the slots with the nozzle axis
- N = number of swirl slots
- A = cross-sectional area of a single swirl slot, in^2

The function f, g, and h for the three spray characteristics D, θ, and S were presented in curve form in the referenced report. By use of the curves of f_d, g_d, and h_d, the following expression was derived for D:

$$D = 286\,(d + 0.17)\,(e^{13/V_v})\,(e^{-0.0004Vt}) \tag{4}$$

A similar expression for the spray angle, θ, was not developed.

By utilization of the expression for flow through an orifice,

$$Q = C\,\pi/4\,d^2\,\sqrt{P} \tag{5}$$

where

Q = orifice flow rate
C = discharge coefficient
d = orifice diameter
P = orifice pressure drop

the pressure drop through the nozzle can be determined for a given flow rate, and a given orifice diameter.

It has been shown (Ref. 2) that the discharge coefficient, C, for a swirl type nozzle is a function of nozzle geometry only, and not a function of injection pressure. By utilization of this result, it can be seen from equation (5) that the flow rate for a given swirl nozzle is a function of pressure drop only. An expression for the discharge coefficient for a swirl type nozzle has been developed, it is presented below:

$$C = 1.17\,\sqrt{(1-\phi)^3/(1-\phi)} \tag{6}$$

where

ϕ = the square of the ratio of diameter of the air core in the outlet orifice to the diameter of the outlet orifice.

The nozzle dimensions are related to the term ϕ by the equation,

$$K = NA/dd_s = \sqrt{(1-\phi)^3/2\phi^2} \tag{7}$$

Where N, A and d are as defined above and d_s is the diameter of the swirl chamber. By use of equations (6) and (7), the discharge coefficient for a given set of nozzle dimensions can be determined.

The design curves presented by Tate and Marshall and equations 2, 5, 6, and 7 were used in the design of the dual orifice nozzle and the spill nozzle. The procedure followed during the study is outlined in Section VIII.

b. REFERENCES

1. Tate, R. W. and Marshall, W. R., "Atomization by Centrifugal Pressure Nozzles", Chemical Engineering Progress, May 1953.
2. Griffen, E. and Muraszew, A., The Atomization of Liquid Fuels, John Wiley & Sons, New York, 1953.

2. THE IMPINGEMENT NOZZLE

a. THEORY

In the design of the impingement nozzle an empirical equation that was developed by Yasusi Tanasawa (Ref. 3) was used. The following equation relates the droplet size, D, to orifice diameter, d, impingement velocity, V_i, surface tension, σ, and liquid density, ρ. This equation is

$$D = 1.73\, d^{0.75}\, V_i^{-0.5}\, (\sigma g_c/\rho)^{0.25}$$

In this equation the velocity V_i, is the impingement velocity. For a 90 degree impingement, this velocity would be the jet velocity. For a jet whose axis makes an angle, θ, with the impingement surface, the equation can be written in terms of the jet velocity, V, by using in place of V_i, the term $V \sin \theta$. Furthermore the equation can be written in terms of the velocity at the outlet orifice by use of the expressions,

$$V = 8\, V_o d/h, \quad h = a/\sin \theta,$$

$$\text{and } V_o = Q/(\pi d^2/_4)$$

In these expressions,

V_o = velocity of the jet in the outlet orifice
d = diameter of the outlet orifice
h = distance along the axis of the jet from the outlet orifice to the impingement surface
a = perpendicular distance from the impingement surface to the outlet orifice diameter
Q = volumetric flow rate

The first of these expressions is the equation for the velocity of a free jet at a distance h from the orifice. (This equation is derived in Reference 4.) The second expression is derived from a consideration of geometry. The last expression is simply the expression of the velocity in a circular orifice. When these expressions are substituted into Tanasawa's equation, it becomes

$$D = 1.73\, [32\, Q \sin^2\theta/a\, \pi\, d^{2.50}]^{-0.5}\, [\sigma g_c/\rho]^{0.25}$$

It should be noted any consistent set of units can be used for the terms in this equation.

In the analytical nozzle design study the droplet size was computed by use of the last equation for a given orifice size, and a given flow rate. An impingement angle of 45° and the distance from the impingement surface was assumed to be 1.0 inch. The fuel properties were known.

b. REFERENCES

3. Tanasawa, Yasusi, "The Atomization of Liquid by Means of Flat Impingement," ARS Journal (paper presented at the American Rocket Society 11th Annual Meeting, November 26-30, 1956).

4. Prandtl, Ludwig, Essentials of Fluid Dynamics, Hafner Publishing Company, New York, 1952, pages 121-123.

3. THE AIR ASSIST NOZZLE

a. THEORY

The air requirements required to atomize the fuel in the 34 to 240 lb/hr range were computed, by an empirical expression presented in Ref. 5. This expression is

$$D = 586\sqrt{\sigma}/V\sqrt{\rho} + 597\,(\mu/\sqrt{\sigma\rho})^{0.45}\,(1000\,Q_L/Q_A)^{1.5}$$

In this equation,

D	=	mean droplet diameter in microns
Q_L, Q_A	=	volume rates of flow of liquid and gas respectively at the discharge conditions,
σ	=	surface tension in dynes/cm,
ρ	=	liquid density in grams/cm^3,
μ	=	liquid viscosity in poises,
V	=	the air velocity at the nozzle exit in m/sec.

The air velocity required to produce a spray with a certain mean droplet for a given flow rate, a given fuel and a certain air orifice size. It should be noted that in the study a 1 inch air orifice diameter was assumed. With this orifice size, the required air velocity, and the air density, the air mass flow and the energy requirements can be calculated. In the study values of air density were used that correspond to actual combustor conditions.

b. REFERENCE

5. Lewis, B., Pease, R. N., and Taylor, H. S. (Editors), Combustion Processes, Volume II of High Speed Aerodynamics and Jet Propulsion, Princeton University Press, 1956, p. 412.